Copyright, 1922, by the American Society of Naval Engineers.

JOURNAL
OF THE
AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXXIV. AUGUST 1922. No. 3.

The Society as a body is not responsible for statements made by individual members.

COUNCIL OF THE SOCIETY
(Under whose supervision this number is published)

Rear Admiral C. W. Dyson, U.S.N. — Rear Admiral B. C. Bryan, U.S.N.
Captain Robert Stocker, (C.C.) U.S.N. — Commander S. C. Hooper, U.S.N.
Captain E. C. Kalbfus, U.S.N. — Lieutenant M. R. Daniels, U.S.C.G.
Commander S. M. Robinson, U.S.N.

WAR-TIME DESTROYER PROGRAM.

By Lieut.-Comdrs. W. C. Owen
and
John F. Shafroth, Jr., U. S. Navy, Members.

The war-time destroyer program was, in accordance with various acts of Congress, passed between August 29, 1916, and July 1, 1918; the destroyers authorized are listed in the following table.

All of these destroyers are now completed or nearing completion, with the exception of destroyers Nos. 200-205, the contracts for which were cancelled after the armistice was signed. Twelve more destroyers were authorized by the act of July 1, 1918, but the contracts have not been let.

Destroyers No.	Builder	Date Authorized	Date of Contract
75–78	Bath Iron Works	Aug. 29, 16	Nov. 30, 16
79–86	Fore River Shipbuilding Co.	Aug. 29, 16	Dec. 26, 16
87–92	Union Iron Works	Aug. 29, 16	Dec. 30, 16
93–94	Navy Yard, Mare Island	Aug. 29, 16	Dec. 6, 16
95–102	Fore River Shipbuilding Co.	Mar. 4, 17	Apr. 27, 17
103–112	Union Iron Works	Mar. 4, 17	May 4, 17
113–118	Cramp Ship & Engine Bldg. Co.	Mar. 4, 17	May 2, 17
119–124	Newport News D. D. & Shipbuilding Co.	Mar. 4, 17	June 29, 17
125–130	New York Shipbuilding Co.	Mar. 4, 17	July 11, 17
131–134	Bath Iron Works	Mar. 4, 17	Aug. 31, 17
135	Navy Yard, Charleston	Mar. 4, 17	Aug. 7, 17
136–141	Navy Yard, Mare Island	Mar. 4, 17	July 20, 17
142–156	Cramp Ship & Engine Bldg. Co.	Mar. 4, 17	Sept. 8, 17
157–160	New York Shipbuilding Co.	Mar. 4, 17	Sept. 28, 17
161–170	Fore River Shipbuilding Co.	Mar. 4, 17	Sept. 7, 17
171–180	Union Iron Works	Mar. 4, 17	Aug. 15, 17
181–185	Newport News D. D. & Shipbuilding Co.	Mar. 4, 17	Sept. 28, 17
186–205	Newport News D. D. & Shipbuilding Co.	Oct. 6, 17	Feb. 14, 18
206–230	Cramp Ship & Engine Bldg. Co.	Oct. 6, 17	Dec. 19, 17
231–250	New York Shipbuilding Co.	Oct. 6, 17	Dec. 29, 17
251–295	Fore River Shipbuilding Co.	Oct. 6, 17	Dec. 6, 17
296–335	Union Iron Works	Oct. 6, 17	Dec. 6, 17
336–341	Navy Yard, Mare Island	Oct. 6, 17	Jan. 22, 18
342–344	Navy Yard, Norfolk	Oct. 6, 17	Jan. 25, 18
345–347	Bath Iron Works	July 1, 18	Jul. 27, 18

No attempt will be made in this article to give the exact description of any particular destroyer, but rather the installation of the destroyer as a general type, and notes made where certain groups, in many cases, vary from this installation.

PRINCIPAL HULL DIMENSIONS.

Length between perpendiculars, feet and inches	310– 0
on L. W. L., feet and inches	310– 0
over all, feet and inches	314– 4½
Breadth, extreme, on L. W. L., feet and inches	30– 8
molded, feet and inches	30–11½
Draft, mean, to L. W. L., feet and inches	9–0 to 9–4
Displacement, corresponding, tons	1154 to 1215
per inch at L.W.L., tons	15.4 to 15.5
Ratio of length to beam (L.W.L.)	10.02 to 1
Coefficient of block	.472
midship	.754
L.W.L. plane	.680

GENERAL DESCRIPTION OF HULL.

All the destroyers of this program are of the flush deck type and have four smoke pipes. The earlier boats had two wooden masts and a twenty-four inch searchlight on the top of the emergency cabin forward of the foremast, and a similar light on the after deck house abaft the mainmast. These boats also were provided with gun foundations for only four-inch 50-caliber guns.

Experience gained during the war indicated that the above design was susceptible of improvement, and accordingly the later boats of the program were provided with gun foundations for five-inch 51-caliber guns, the searchlights were placed on a platform abaft the smoke pipes and so arranged that both or either light could be used on either side, and the tall mainmast was replaced by a small stub mast only high enough to carry the radio aerial well clear of the smoke pipes. These boats were provided with a small navigating searchlight on top of the emergency cabin to replace the twenty-four-inch light moved amidship.

Main Deck.—The main deck extends the entire length of the vessel and slopes aft from stem to stern. The deck at the forward point of the forecastle is sixteen feet above the water-line, while at the stern it is only eight feet.

Chart House.—Radio Room.—The chart house and radio room are on the main deck between frames 40-51, the chart house being located on the center line while the radio room is abaft the chart house and slightly offset to port. There is a passage-way on both sides outboard of the chart house and radio room.

The forward sides of the chart house are extended to the sides of the ship forming a watertight bulkhead which affords protection to the crew in wet weather and keeps the after part of the main deck comparatively free from water. This bulkhead has a watertight door on each side and these afford access to and from the forecastle.

Bridge.—The top of the chart house and the radio room forms part of the deck of the navigating bridge. The after end of the bridge extends from side to side of the ship and is just forward of the foremast. The forward side of the bridge conforms in shape to the watertight bulkhead and forward sides of the chart house, and a comfortable and roomy bridge is obtained. The bridge is provided with parabolic wind-shields and is enclosed with non-shatterable glass windows, generally of the vertical sliding type. On most vessels, an emergency cabin of comfortable size is located on the center-line near the after end of the bridge. The top of this emergency cabin forms the main fire control and director station of the ship.

Midship Deck House.—Between the second and third smoke pipes is built up a midship deck house, sufficiently strong to carry the two main battery guns mounted thereon. In the earlier boats the sides of this structure were open, but in the later boats where five-inch foundations were necessary the

frames supporting the deck house were covered with steel plate.

Galley.—Inside the superstructure and on the main deck on the centerline is placed the galley, fitted with a two-section oil-burning range, one twenty-gallon coffee urn and two forty-gallon steam-jacketed kettles.

Outside the galley on either side are the trunks for the forced draft blowers that supply air to the firerooms, while outboard of these, on either side of the ship is a passage-way.

After Deck House.—The after deck house extends from frame No. 144 to frame No. 155 and includes the crew's wash-room and toilets, and the torpedo repair shop.

First Platform Deck.—From forward to aft are located stores, anchor engine room, C. P. O.'s quarters, showers and toilet, officers' quarters, office, and abaft the engine room, crew's quarters and steering engine room.

Second Platform Deck.—This deck extends from the bow to frame No. 49, and on it are located from forward to aft, paint locker, chain locker, store-rooms and crew's quarters.

COMPLEMENT.

Provision is made for the following complement:

Commanding Officer	1
Other Officers	5
Crew	110
Total	116

BATTERY.

The battery of these destroyers consists of four four-inch fifty-caliber case guns and one three-inch twenty-three caliber anti-aircraft gun. The main battery is placed as follows:

One gun on the forecastle amidships.

One gun on starboard side of the Midship Deck House.

One gun on port side of the Midship Deck House.
One gun on top of after deck house.

In many destroyers this last gun was on the main deck abaft the deck house, but this location was too low and too wet for effective use of the gun, and all the destroyers having it so placed are being altered so as to have the after gun on top of the deck house. The anti-aircraft gun is so located as to allow it to cover the maximum area of sky; its location is not uniform on the vessels; and while it is usually on the forecastle abaft the four-inch gun, some vessels have it fitted on the top of the after deck house and other vessels have it located on the after part of the main deck. As previously stated, the later boats are provided with gun foundations for five-inch guns, though four-inch guns are actually installed on all vessels except Nos. 231 to 235, inclusive, on which vessels the five-inch guns were fitted.

TORPEDOES.

There are four triple twenty-one-inch tubes placed two on either side, and a Bliss Levitt torpedo is provided for each tube, each destroyer thus carrying twelve torpedoes.

SMALL BOATS CARRIED.

The following small boats are allowed to each vessel:
One 21-foot motor dory.
One 24-foot motor sailing launch.
One 24-foot whale boat.
One 14-foot wherry.
One 10-foot punt.

The motor dory and motor sailer are stowed in cradles placed on skid beams about seven feet above the main deck. The whale boat is hung at its davits and the punt and wherry are stowed on the main deck.

ANCHOR GEAR.

Each vessel carries two navy type anchors weighing 1800 pounds each, and 105 fathoms of chain is provided for each anchor. A trial installation of hawse pipes in connection with a flush type of stockless anchor has recently been completed on the *Truxton* (D. D. 229). The anchor windlass is a two-cylinder 5×5 vertical double-acting steam engine.

STEERING GEAR.

The steering engine is located aft on the first platform deck, in compartment D-208. There are two steering stations with connections to the steering engine for steam steering, one on the bridge and the other on the deck house, aft. A wheel is located at the steering engine for steering, and hand steering wheels are provided in the steering engine-room.

DRAINAGE SYSTEM.

The fire and engine rooms are drained by a main drain which extends from the after end of the engine-room to the forward end of the forward fire-room. There is a suction to the main drain at the forward and after end of each fire and engine room. There is a manifold for pumping the forward compartments in the forward fire-room, and one for pumping the after compartments in the after engine-room.

FIRE MAIN.

The fire main is supplied by four fire and bilge pumps, one located in each engine-room and one in each boiler-room. It is a three-inch lead-lined steel tubing and extends continuously through the entire length of the machinery spaces directly under the main deck beams, and then continues forward and aft when it leaves the machinery spaces.

FLUSHING SYSTEM.

Branches with reducing valves are taken from the fire main for supplying water to the officers' water-closet, crew's water-closets and urinals, refrigerating machine for cooling purposes, and to the faucets in the crew's washroom aft.

VENTILATING SYSTEM.

The living quarters, steering engine room and radio-room are provided with forced ventilation. On the vessels now being completed, a ventilation blower is fitted in a ventilator discharging into engine-room. This is also being fitted on some of the vessels already completed.

DESCRIPTION OF MACHINERY.

MAIN ENGINE.

The vessels are all twin screw and all turbine driven, with five different types of main propulsive units installed as follows:

Parsons geared turbine.
Fore River Curtis geared turbine.
General Electric Curtis geared turbine.
Westinghouse geared turbine.
Newport News Curtis turbine direct drive, with one geared cruising turbine.

The two main units, with their independent condensing plants, etc., are installed in separate engine rooms; the port unit being the forward one.

Parsons Geared Turbines.—Parsons geared turbines are installed in 85 of the vessels, as follows: 75, 78, 93 and 94, 113, 118, 125, 160, 206-230 and 336-347. The original design, as fitted in the earlier of these groups, was for 12,100 S.H.P. per unit or 24,200 S.H.P. per vessel, with steam at

240 pounds gauge at the H.P. turbine and the propellers making 448 R.P.M. In the later vessels having greater oil carrying capacity and the consequent increase in displacement, this power was increased to around 27,000 S.H.P. per vessel; the greater power being obtained by slightly lengthening the blades of the first three full power stages and increasing the steam pressure to 250 pounds at the H.P. turbine.

Each propeller shaft is driven through single reduction, double helical gearing, by one single-flow high pressure turbine and one double-flow low pressure turbine, each turbine driving through its individual pinion; the two pinions connect with the single gear on the same horizontal plane, being located diametrically opposite each other.

A single flow astern turbine is fitted in the forward end of the low pressure ahead turbine casing; the two astern turbines being capable of driving the vessel astern at 20 knots.

For full speed, steam is admitted into the main belt (5th stage) of the H.P. turbine, expands through the remaining stages of this turbine, then through the L.P. turbine and to the condenser. For cruising speeds, the steam is admitted into the low cruising belt (1st stage) or into the higher cruising

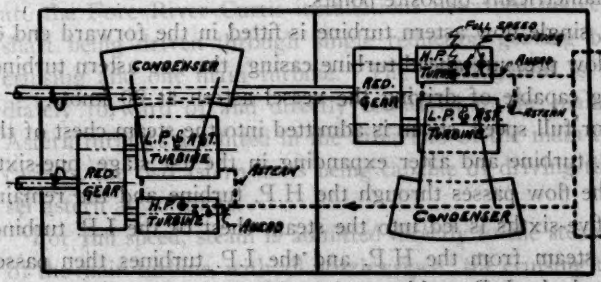


FIG. 1.—PARSONS GEARED TURBINE, FITTED ON DESTROYERS NOS. 75 TO 78, 93 AND 94, 113-118, 125-160, 206-230, 336-347.

belt (3rd stage) of the H.P. turbine and carried through the H.P. turbine, then through the L.P. turbine and into the condenser.

The blading is of the reaction type throughout, and the designed speeds are 2988 R.P.M. for the H.P., and 1775 R.P.M. for the L.P. turbine.

Fore River Curtis Geared Turbine.—Fore River Curtis turbines with Falk reduction gearing, are installed in 91 of the vessels, as follows: Nos. 79-86, 95-112, 161-180 and 251-295. These turbines are designed for 13,500 S.H.P. per unit, making 27,000 S.H.P. per vessel, with steam at 250 pounds gauge at the H.P. turbine and the propellers making 452 R.P.M.

Each propeller shaft is driven through single reduction, double helical gearing, by one high pressure turbine, one intermediate pressure turbine and one double-flow low pressure turbine. The intermediate and low pressure turbines each connect to a pinion shaft through a flexible coupling, the high pressure turbine being forward of and connected with the intermediate pressure turbine by means of a similar flexible coupling. The two pinions, as in the Parsons installations, are in the same horizontal plane and connect with the single gear at diametrically opposite points.

A single flow astern turbine is fitted in the forward end of the low pressure ahead turbine casing; the two astern turbines being capable of driving the vessel astern at 20 knots.

For full speed, steam is admitted into the steam chest of the H.P. turbine and after expanding in the 1st stage, one-sixth of the flow passes through the H.P. turbine and the remaining five-sixths is led into the steam chest of the I.P. turbine. The steam from the H.P. and the I.P. turbines then passes through the L.P. turbine and to the condenser. For cruising speeds, steam enters the H.P. turbine steam chest and passes through all three turbines to the condenser.

The designed speeds are 3230 R.P.M. for the H.P. and the I.P. turbines, and 3000 R.P.M. for the L.P. turbine. Particulars of the gears will be considered in a later paragraph.

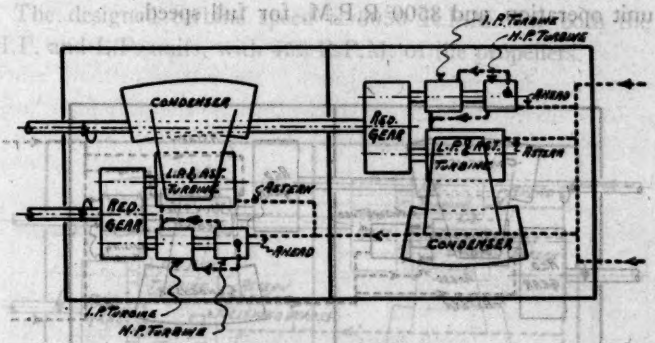


FIG. 2.—FORE RIVER CURTIS GEARED TURBINE, FITTED ON DESTROYERS NOS. 79-86, 95-112, 161-180 AND 251-295.

General Electric Curtis Geared Turbine.—This make of propelling unit was installed in 46 of the vessels, viz., Nos. 87 to 92 and 296 to 335, built at the San Francisco plant of the Bethlehem Shipbuilding Corporation. These units are designed for the same power and propeller shaft revolutions as are the Fore River Curtis installations described above, each shaft being driven through single reduction gearing by one cruising and one main turbine. The cruising unit is immediately forward of and directly connected to the main unit. Astern turbines are fitted in the after end of the main turbine casings, the two astern units being capable of driving the vessel astern at 20 knots.

For full speed, steam is admitted directly to the steam chest of the main turbine, with all nozzles open; the cruising turbine running in a vacuum.

For the higher cruising speeds the same condition prevails except only a limited number of steam nozzles are open.

For cruising speeds up to 20 knots, steam is admitted to the cruising turbine, passes through that turbine, then through the main turbine and to the condenser.

The designed turbine speeds are 1856 R.P.M. for cruising unit operation, and 3500 R.P.M. for full speed.

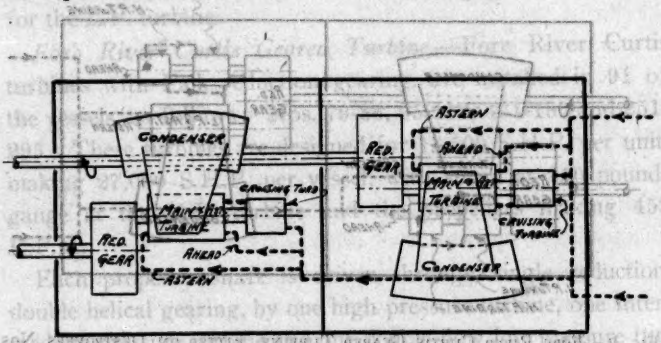


FIG. 3.—GENERAL ELECTRIC CURTIS GEARED TURBINE, FITTED ON DESTROYERS NOS. 87-92 AND 296-335.

Westinghouse Geared Turbines.—Westinghouse Turbines and Reduction Gears were installed in 34 of the vessels, viz., Nos. 186-199 and 231 to 250. These units are designed for 13,500 S.H.P. each, as in the latest Parson's geared installations, and the arrangement of the unit is also the same. The first stage blading of the H.P. turbine, however, is of the impulse type as is also the case with the astern turbine blading.

For full speed, steam is admitted to the 1st stage (impulse blading) of the H.P. turbine, by passes the cruising stages (the first 3 stages of reaction blading), expands through the remaining stages of this turbine, then through the L.P. turbine and to the condenser.

For the higher cruising speeds, the steam follows the same course, excepting that a fewer number of nozzles on the H.P. steam chest are opened.

For the lower cruising speeds, steam is admitted into the first or impulse stage blading of the H. P. turbine through a still fewer number of nozzles, expands through all stages of this turbine, then through the L.P. turbine to the condenser.

The designed turbine speed is 3050 R.P.M. for both the H.P. and L.P. units, with 452 R.P.M. of the propellers.

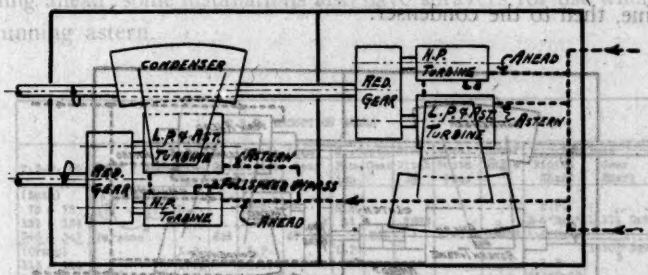


FIG. 4.—WESTINGHOUSE GEARED TURBINES, FITTED ON DESTROYERS NOS. 186-199 AND 231-250.

Newport News Curtis Turbine.—This type of turbine was installed on 11 vessels, viz., Nos. 119-124 and 181 to 185. The main turbines are direct connected to the propeller shafting, and one geared cruising turbine is installed and connected by means of a jaw clutch to the forward end of the starboard main turbine. This cruising unit is disconnected at speeds over 20 knots; when in service, the cruising unit exhausts into the port main turbine.

These installations are designed for a total of 25,000 S.H.P. per vessel, with steam at 250 pounds gauge at the main throttles and the propellers making 625 R.P.M.

An astern turbine is fitted in the after end of each main turbine, the two astern units being capable of driving the vessel astern at a speed of 20 knots.

For full speed, steam is admitted into the steam chests of the main turbines and all nozzles opened.

For cruising speeds over about 20 knots, steam is still admitted to the H.P. turbine steam chest, but with the number of open nozzles reduced to suit.

For the lower cruising speeds, steam is admitted to the geared cruising turbine connected to the starboard main unit; as noted above, the exhaust passes through the port main turbine, then to the condenser.

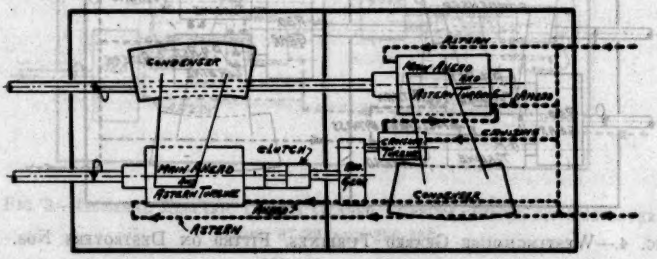


FIG. 5.—NEWPORT NEWS CURTIS TURBINES, FITTED ON DESTROYERS NOS. 119-124 AND 181-185.

REDUCTION GEARS.

All of the gears are of the single reduction, double helical type and, except in the case of the General Electric Company's gears, the pinions mesh directly with the gear wheel. In the General Electric Company's type, the ratio of pinion and gear is higher than on the other installations and the pinion meshes with two idlers, both of which are in contact with the gear; the arrangement being as shown in the tabulated data below.

The gears and pinions are all fully enclosed in a single casing and aside from the Westinghouse units, they are mounted on a common frame, with the bearings rigidly fixed. The pinions of the Westinghouse gear installations are mounted on the flexible or "floating pinion frame" peculiar to that make of gears.

The pinion shafts connect with the turbine shafts by means of flexible couplings while the gear shaft is provided with a flanged coupling on its after end to match the line shaft to which it is securely bolted.

All gears and bearings are plentifully supplied with oil from the forced lubricating system, the gear teeth being lubricated by a series of spray nozzles delivering oil to the point where the driving teeth enter in engagement when the gear is running ahead; some installations also have sprayers for use when running astern.

T.B.D. REDUCTION GEARS

T.B.D. No.	Type & Make	R.P.M. Pinion	R.P.M. Gear	Pitch Diam. Pinion	Pitch Diam. Gear	No. Teeth Pinion	No. Teeth Gear	Circ. Pitch	Total Tooth Face	Spiral Angle	Bearings Pinion Shaft	Bearings Gear Shaft
(Bath) 75 - 78 131 - 134 245 - 247	L.P. H.P. Parsons	H.P. 2908	436	H.P. 9.916"	67.962	H.P. 41	281	.7896"	27-3/4"	25-53°	H.P. 2-6"Dx6" 1-6"Dx11"	1-11½" Dx9"
(Cramp) 115 - 118 142 - 156	(Out by DeLaval Co.)	L.P. 1778		L.P. 16.69"		L.P. 49					L.P. 2-8"Dx8" 1-8"Dx11"	1-10⅝" Dx9"
(N.Y.Ship) 125 - 130 157 - 162 (Norfolk) 242 - 244											(5-1/2% nickel Steel forg.)	cast steel center, spcl. forg steel rim
(Mare Isl) 93 & 94 136 - 141 224 - 241 (Ch'ston) 135	" (Out by Mare Isl.	H.P. 2977 L.P. 1769	"	"	67.721	"	280	"	"	"	"	"
(Fore R.) 79 - 86 96 - 102 161 - 170 251 - 255 (U.I.Wks) 103 - 112 171 - 180	L.P. H.P. "Falk"	H.P. 3230 L.P. 2000	458	H.P. 10.06" L.P. 11.6"	77.8"	H.P. 37 L.P. 29	198	1.256"	35	44-58°	2-8"Dx12" 2-8"Dx11" (chrome nickel steel forg.)	1-14"Dx15" 1-14"Dx12" cast steel center, cl.D forg steel rim
(N. Mass) 126 - 129 (N.Y.Ship) 221 - 250	L.P. H.P. Westinghouse	3000	452	HP & LP 10.227	66.312	51	201	1.035"	24"	32°	2-8"Dx7" 1-8"Dx9" (West. spcl. forged steel)	2-11.26" D.x10" Cast iron center, cast steel rim
(U.I.Wks) 67 - 92 226 - 236	Gen.Elec.Co.	3000	452	8.888 (Idlers 26.167)	68.328	38 (Idl) 197	410	.8884	21-1/4"	27°	2-7"Dx10" (Idlers 2-10"Dx20")	2-14"Dx18" CI.center spcl P.S. rim.

* Measured circumferentially (not at right angle to teeth)

SHAFTING AND PROPELLERS.

The shafting on all of the vessels is of nickel steel, finished all over and with axial holes bored from end to end of each length of shaft. Each line of shafting consists of a propeller shaft, a stern tube shaft and one length of line shafting, with an additional length of line shafting for the port side extending to the forward engine room.

The propeller and stern tube shafts are connected by an outboard coupling, consisting of a sleeve fitted to the tapered ends of the shafts and secured by cross keys and feathers.

The stern tube shaft is arranged for withdrawal outwards through the stern tube and is connected with the flanged coupling of the adjacent line shaft by a coupling consisting of a sleeve and a thrust collar in halves, to prevent the shaft from pulling out when backing.

The body of each propeller shaft and each stern tube shaft is covered by sleeves at the bearings, the sleeves being shrunk on the shaft and pinned thereto. The stern tube shafts are also covered with a light water-tight casing between the stern tube bearings.

The stern tube and strut bearings consist of strips of lignum-vitae, well fitted into composition bearing bushings.

The propellers are all true screw, three bladed, and of manganese bronze, with the blades cast solid with the hub; the hub is accurately fitted to the taper end of the propeller shaft, keyed and held in place by a locked nut. The starboard propeller is right hand and the port propeller left hand.

The hubs and caps are finished all over, machined where possible and polished. The driving face of each blade is machined to a true surface and polished, the backs of the blades being also polished.

The matter of interchangeability of the propellers and propeller shafts for these vessels was carefully considered, with the result that but three types of propeller shaft and of pro-

peller hub have been fitted; 105 of the vessels are provided with propeller shafts and propellers of the so-called Bath type, and 151 with the Fore River type. The 11 Newport News direct drive installations comprise the third interchangeable group. The general propeller data is given in the following list:

T.B.D. PROPELLERS

T.B.D. No.	Dia. Ins.	Pitch Ins.	R.P.M.	P.a. sq.ft.	P.d. Ratio	Dia. Shaft	Length Hub	Remarks
Bureau Eng. Type Plan	108	119	445	37.87	.6	11 1/2"	29"	
(Bath) 75 - 79 131 - 134 245 - 247 (Cramp) 113 - 116 142 - 186 208 - 209 (Mare Isl.) 93 & 94 136 - 141 254 - 261 (Ch'eston) 122 (Norfolk) 242 - 244 (N.Y.Ship) 125 - 130 167 - 180	110	122	445	36.29	.68	11 1/4"	33 1/4"	
221 - 230	108	119	452	37.66	.698	"	"	Interchangeable with the above vessels.
(Fore R.) 79 - 86 95 - 102 161 - 170 251 - 258 (U.Wks) 87 - 92 103 - 112 171 - 180 259 - 266	108	119	445	37.8	.595	11 1/2"	29"	
(N.News) 186 -196	"	"	450	36.28	.601	"	"	Interchangeable with F.R. and U.I.Wks vessels.
(N.News) 116 - 126 181 - 185	98½/8	89	488	30.68	.648	10 3/4"	29"	

The main thrust bearings are of the pivoted, segmental type for all of the vessels. For the geared turbine installations the thrusts are either 21-inch or 23-inch Kingsbury bearings, and are located on the forward end of the reduction gear slow speed shaft, being carried in a casing forming an extension of the gear case.

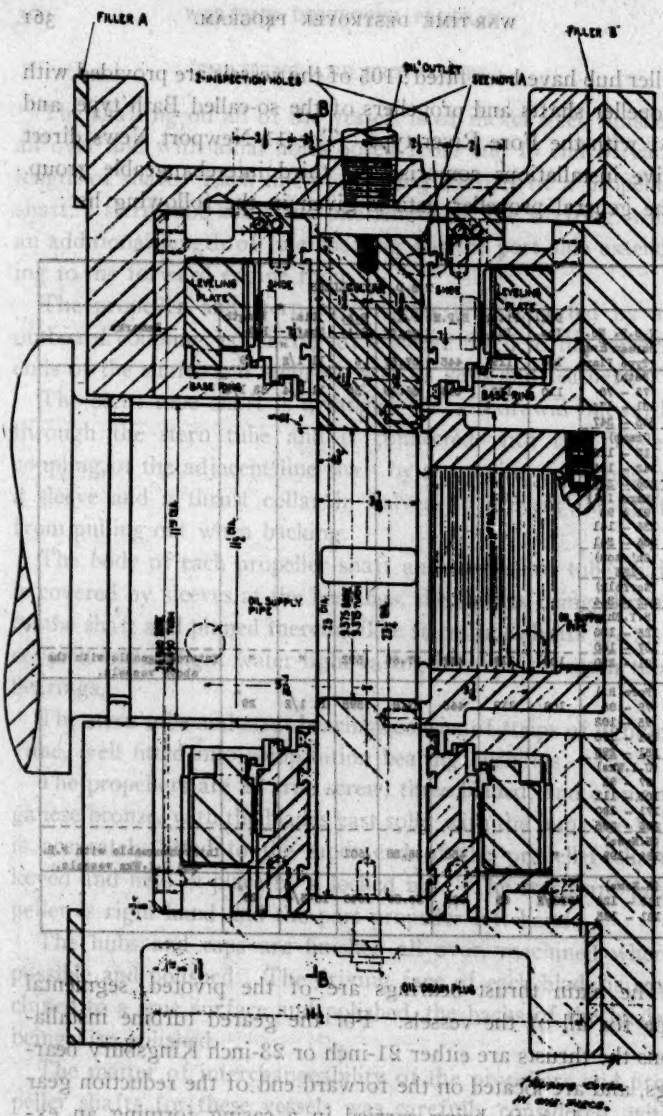

FILLER A
FILLER B
2-INSPECTION HOLES
OIL OUTLET
SEE NOTE A
LEVELING PLATE
SHIM
BASE RING
OIL SUPPLY PIPE
OIL DRAIN PLUG

FIG. 6.

For the direct drive installations, the main thrust bearing is combined with the turbine thrust bearing, and is located at the forward end of the main turbine. These thrusts are 16-inch Kingsbury type.

In this connection it should be stated that with the exception of the General Electric Company installations, all of the turbine thrust bearings on these vessels are of the Kingsbury type, the diameter of collar varying from 7 inches to 9 inches.

Fig. 6 is a sectional view of a 23-inch Kingsbury thrust bearing as installed on a number of the vessels.

The lists below show the diameter of the thrust collars and the corresponding number of shoes with their surface, both for the main and the turbine thrust bearings for the various groups of vessels.

MAIN CONDENSING APPARATUS.

An independent condensing outfit is installed for each of the two main propelling units, on all of these Destroyers.

A surface condenser, with a cooling surface ranging from 7000 square feet to 7600 square feet, and arranged for scoop circulating water supply, is provided in each engine room. The condenser shells are of steel plate, the tube sheets of rolled naval brass and the circulating water ends, with few exceptions, are of copper as are the main injection and overboard delivery pipes. The tubes are expanded into the tube sheets at both ends, and are bent to a large radius to permit proper expansion.

An air pump, and also a circulating pump for use when getting under way, are provided for each condenser. The air pump is arranged for both wet and dry suction on a number of the installations, all other vessels being provided with augmenters and augmenter condensers.

T.B.D. TURBINE THRUST BEARINGS.

T.B.D. NO.	TYPE	R.P.M.	COLLAR Diam. Ins.	COLLAR Thick Ins.	RINGS I.D. Ins.	RINGS O.D. Ins.	No. of RINGS Ahead	No. of RINGS Ast.	AREA SURFACE Ahead Sq. Ins.	AREA SURFACE Astern Sq. Ins.
(Bath) 74	Kingsbury	(H.P. 2562 / L.P. 1487)	6 1/8	1	3	6	8	8	9	9
(Bath) 75 - 78 131 - 134 345 - 347	"	(H.P. 2988 / L.P. 1775)	9 1/8	1 3/8	4 1/4	9	8	8	24.5	24.5
(Cramp) 115 - 118 142 - 156 206 - 220	"	"	"	"	"	"	"	"	23.56	23.56
(Mare Isl) 93 & 94 136 - 141 336 - 341	"	"	"	"	"	"	"	"	24.5	24.5
(Chs'ton) 135	"	"	"	"	"	"	"	"	"	"
(Norfolk) 342 - 344	"	"	"	"	"	"	"	"	"	"
(N.Y.Ship) 125 - 130 157 - 160	"	"	"	"	"	"	"	"	28.38	28.38
(Fore R.) 79 - 86 95 - 112	"	H.P. 3230	7 1/8	1	4 1/2	7	6	6	14	14
161 - 170 251 - 295		I.P.&L.P. 3000	8 1/8	1	5 1/2	8 & 8	8	8	18	18
(U.I. Wks) 103 - 112 171 - 180	"	"	"	"	"	"	"	"	"	"
(N. News) 119 - 124 181 - 185	"	(Cr. Parsons) 625	7 1/8	1	4 1/2	7	6	6	15.528	15.528
(N. News) 186 - 199	"	3060	8 1/8	1 3/8	5 3/4	8	6	6	27.77	27.77
(N.Y.Ship) 231 - 250	"	"	"	"	"	"	"	"	"	"
(U.I. Wks) 87 - 92 296 - 335	G.E.Co.	3500	13 1/4	2 1/2	7 5/8	13 1/4	(Solid plate in halves)		81	81

	T.B.D. MAIN THRUST BEARINGS.									
T.B.D. NO.	TYPE	RPM	COLLAR		SHOES		Rubbd Shoes		SHOE SURFACE	
			Diam. Ins/	Thick Ins.	I.D. Ins.	O.D. Ins.	Ahead	Astr.	Ahead sq.ins.	Astr. sq.ins.
(Bath) 74	Kingsbury	462	16 1/2	3	8 1/4	16 1/4	4	4	111.27	111.27
(Bath) 75 - 78 131 - 134 345 - 347	"	450	21 1/4	3 1/4	10 1/2	21	6	6	216.75	216.75
(Cramp) 113 - 118 142 - 166 206 - 230	"	"	"	"	"	"	"	"	"	"
(Mare Isl) 93 & 94 126 - 141 336 - 341	"	"	"	"	"	"	"	"	"	"
(Chs'ton) 135	"	"	"	"	"	"	"	"	"	"
(Norfolk) 342 - 344	"	"	"	"	"	"	"	"	"	"
(N.Y.Ship) 125 - 130 157 - 160	"	"	"	"	"	"	"	"	"	"
(Fore R.) 79 - 86 98 - 102 161 - 170 251 - 295	"	452	23 1/4	3 1/2	11 1/2	23	6	6	260.6	260.6
(U.I.Wks.) 87 - 92 103 - 112 171 - 180 296 - 335	"	"	"	"	"	"	"	"	"	"
(N.Y.Ship) 231 - 250	"	"	"	"	"	"	"	"	"	"
(N.News) 186 - 199	"	"	"	"	"	"	"	"	"	"
(N.News) 119 - 124 181 - 185	"	625	16 1/4	2 1/4	7 1/2	16	6	6	127	127

T.B.D. CONDENSING PLANTS

T.B.D. No.	Main Condenser (1) No. & Type	C.S. of one. Sq.ft.	Tubes	Circ. Water Conn.	Thickness shell	Air Pumps No. Make & Size	Sta.	Dia.	No.	Auxiliary Condenser C.S. of one. (sq.ft)	Circ. Water	Air Pump Dia.	Vacuum Augmenter Inlet	Vacuum Augmenter Outlet
(Bath)														
76 - 78	2-circular,	7012	3696-	22"	1/4"	2-Warren	15"	10"	2	167.2	4"	10 1/2	12"	5-1/2"
131 - 134	Tubes bent		5/8"x	Scoop		11x32x21								
245 - 247	hor. plane,		.049"			twin beam								②
(N.Y.Ship)	Expand in		Thick			(2 steam								
126 - 130	tube sheets					cyl.)								
157 - 160														
251 - 253	"	"	"	"	"	2-Worth'n	"	"	"	"	"	"	"	"
						12x20x24								
						twin beam								
						(2 steam								
						cyl.)								
(Mare Isl)														
93 & 94	"	"	"	"	"	2-Dow	"	"	"	"	"	"	"	"
136 - 141						11x32x21								
244 - 241						twin beam								
(Ch'ston)						(2 steam								
136						cyl.)								
(Cramp)														
113 - 118	"	7025	"	"	"	2-Warren	"	"	1	175	"	6"	2-6"	6"
	(Alt.rows					11x32x21								
	bent opp.					twin beam								
	direction,					(2 steam								
	tube sheets					cyl.)								
142 - 156	parallel)					2-Worth'n								
206 - 250		"	"	"	"	15x30x24	"	"	"	"	"	"	"	"
(Norfolk)						twin beam								
342 - 344						(1 steam								
						cyl.)								
(N.News)														
119 - 124	2-circular,	7514	4147-	24"	"	"	15"	11"	2	170.3	4-1/2	11"	15"	5-1/2"
181 - 185	Tubes bent		5/8"x	Scoop										
186 - 199	hor. plane,		.049"											③
	Expand in		Thick											
	tube sheets													
(Fore R.)														
79 - 86	2-	7596	4443-	"	5/16"	"	10"	10"						
95 - 102	year shape		5/8"x				wet							
161 - 170	tubes bent		.049"											
251 - 252	hor. plane,		Thick				8"							
(U.1.Wks)	Expand in						dry							
87 - 92	Tube													
103 - 112	sheets.													
171 - 180														
294 - 336														

① A 14" Worthington cent.circ.pump (turbine driven) of approx. 5000 gpm & 15 ft.head installed for each condenser.

② Installed to 10 1/2" on augmenter condenser.

③ " " 11" " " " "

Auxiliary Condenser.—One auxiliary condenser of approximately 300 square feet cooling surface, is installed for use when in port or at anchor. A combined air and circulating pump is provided for use with this condenser on all except the Fore River and Union Iron Works built destroyers; those vessels were fitted with a vertical reciprocating air pump and an independent engine driven circulating pump.

FORCED LUBRICATION SYSTEM.

Each main propelling unit is provided with its complete lubricating oil system, the two systems being cross connected. Normal operation is for independent running of each system and reserve pumps have been provided with that end in view; where only one reserve pump has been installed, its connections are such as to permit of its use on either the forward or the aft system.

The lubricating oil pumps take suction from the oil drain tank to which all turbine and reduction gear bearings are led, and discharge through strainers and oil coolers to the main supply line, which feeds all bearings. A 500-gallon oil storage (reserve) tank is provided, as well as oil settling tanks and a motor driven centrifugal oil purifier; all having separate connections to each system.

Thermometers are fitted at each oil cooler inlet and outlet as well as at the outlets from turbine and reduction gear bearings.

The oil coolers are of various makes, all being of ample capacity, with cooling water furnished by their own pumps and with an emergency connection to the fire main.

BOILERS.

There are four water-tube boilers of the small tube type installed in each vessel; they are located in two fire rooms (two boilers in each fire room) immediately forward of the engine rooms. Each boiler has its independent uptake and smoke pipe.

T.B.D. LUB. OIL TANKS, PUMPS, ETC.

T.B.D.No.	Settling Tanks No.	Cap cub	Inlet Ins.	Drain Ins.	Drain Tanks No.	Cap cub	Rod.G Drain	Pump Suc.	Lub. Oil Pumps No.	Make	Size	Oil Cooler Elec.Pumps No.	Make	Size
(Bath) 74	2	200	1 1/2	1 1/2	1	340	4	4 1/2	2	Warren	6x8x12 V.S.	2	Warren	7x7x12 V.S.
(Bath) 75 - 78	"	"	"	"	"	250	"	"	3	"	"	"	"	6x7x8 V.S.
131 - 134					1	366								
245 - 247														
(Cramp) 115 - 118	"	"	2	2	"	194	4 1/2	4	3	"	"	"	"	"
					"	238								
248 - 256	"	"	"	"	"	"	"	"	"	Worth-ington	7x8x12 V.S.	"	Worth-ington	6x8x12 V.S.
204 - 220														
(N.Y.Ship) 125 - 130	"	"	1 1/2	1 1/2	1	250	4	4 1/2	3	Warren	6x8x12 V.S.	2	Warren	6x7x8 V.S.
187 - 190					1	366								
					1	360	6							
221 - 230	"	"	"	"	1	310	4	"	"	Worth-ington	7x8x12 V.S.	"	Worth-ington	8x9x12 V.S.
(Mare Is.) 93 & 94	"	"	"	"	1	250	4	"	"	Warren	6x8x12 V.S.	"	Warren	8x7x8 V.S.
					1	366								
136 - 141	"	"	"	"	"	"	"	"	"	Dow	"	"	Dow	"
234 - 241														
(Ch'ston) 135	"	"	"	"	"	"	"	"	"	"	"	"	"	"
(Norfolk) 242 - 244	"	"	2	2	1	194	4 1/2	4	3	Worth-ington	7x8x12 V.S.	2	Worth-ington	8x9x12 V.S.
					1	238								
(N.News) 119 - 124	1	400	1 1/4	1 1/2	1	177	4	4	3	"	"	2	"	"
191 - 198														
196 - 199	2	207	1 1/2	2/4	2	250	6	6	"	"	"	"	"	"
(Fore R.) 79 - 86	2	225	3	2	2	340	6	4 1/2	(Gear-ed) 4	"	"	2	"	"
98 - 102														
(U.I.Wks) 87 - 92	"	"	"	"	"	"	"	"	"	"	"	"	"	"
103 - 113														
(Fore R.) 161 - 170	"	"	2	2	1	340	"	"	"	"	"	"	"	"
251 - 255					1	350								
(U.I.Wks) 171 - 180	"	"	"	"	"	"	"	"	"	"	"	"	"	"
256 - 265														

T.B.D. LUB. OIL COOLERS.

T.B.D. NO.	NO.	TYPE Hor. or Vert.		Oil flow thro-	G.S. of one Sq.ft	No. Dia. Tubes per Cooler	Oil Conn. Ins.	Circ. Water Conn. Ins.	Size Shell I.D. Ins.	Shell Length bet. T.S.	
(Bath) 74	2	H.	Bath #11 2 pass	Tubes	327	287-5/8" (retarders fitted)	4	2 1/2	20	42	
(Bath) 75 - 78 181 - 184 245 - 247	"	"	"	"	"	"	"	"	"	"	
(Cramp) 113 - 118 142 - 156 206 - 230	"	"	"	"	"	"	3 1/2	3 1/2	"	"	
(N.Y.Ship) 125 - 130 157 - 160 251 - 260	"	"	"	"	"	"	4	"	"	"	
(Mare Isl) 93 & 94 136 - 141 336 - 341	"	"	"	"	"	"	"	"	"	"	
(Ch'ston) 135	"	"	"	"	"	"	"	"	"	"	
(Norfolk) 342 - 344	"	"	"	"	"	"	"	"	"	"	
(N. News) 119 - 124 161 - 165	"	"	#109 Reilly Multiwhirl	Shell (water 2 pass)	126	172-5/8"	2 1/2	2 1/2	13 5/8	54	
186 - 199	2	H.	#116 Reilly Multiwhirl	"	300	336-5/8"	4	4	18 1/4	66	
(Fore R.) 79 - 86 95 - 102	"	"	#8 Z Schmitz & Koerting	Shell (water 1 pass)	278	435-9/16"	3 1/2	4	16 7/8	57	
(U.I. Wks) 87 - 92 103 - 112	"	"	"	"	"	"	"	"	"	"	
(Fore R.) 161 - 170 251 - 295	"	"	Reilly Multiwhirl	Shell (water 1 pass)	285	396-5/8"	4	4	19 11/16	52	
(U.I. Wks) 171 - 180 296 - 335	4	"	"	"	"	"	"	"	"	"	

The boilers are all arranged for the burning of oil fuel exclusively, with the fire rooms under an air pressure corresponding to 6 inches or 7 inches of water.

The types of boilers installed on the various groups of vessels, with the corresponding heating surface, size of tubes, etc., are as given in the following list:

	T.B.D. BOILERS. (4 per vessel).												
T.B.D. No.	Type	H.S. for Boiler sq.ft.	Press Gage	Fur. Vol. cu.ft	Fur. Vol / H.S.	I.D.& Thick. Steam Drum	I.D.& Thick. Water Drums	TUBES No.	Dia.	thick ness	No. F.C. Rms	Down Comer Pipes No.& D.
(Bath)												
75 - 78	Normand	6750	260	716	.1085	48"	21"	1096	1 1/4"	.109"	14	8 -
151 - 154						3/4" &	1 1/2" &	288	1 1/2	.120		10"
345 - 347						1 3/8"	7/8"	272	1 3/4	.134		
(Mare Isl)												
93 & 94	"	"	"	"	"	"	"	"	"	"	12	"
136 - 141												
236 - 241											(4	
(Cha'ton)												
135	"	"	"	"	"	"	"	"	"	"	14	"
(Norfolk)												
342 - 344	"	"	"	"	"	"	"	"	"	"	"	"
(Cramp)												
113 - 116	White	6875	265	-		54"	24"	3744	1"	.109"	14	8 -
242 - 246	Forster					3/4" &	9/16 &					4"
306 - 330						2"	1 1/4"					
(N.News)												
186 - 199	"	"	"			"	"	"	"	"	"	"
(N.Y.Ship)												
231 - 250	"	"	"			"	"	"	"	"	"	"
(Fore R)												
79 - 86	Yarrow	6885	256	800	.1162	50"	30"	230	1 1/4"	.109"	12	none
95 - 102						11/16	27/32	3522	1	.095		
161 - 170						&	&					
251 - 295						1 9/16"	1 1/4"					
(U.I.Wks)												
87 - 92												
103 - 112	"	"	"	"	"	"	"	"	"	"	"	"
171 - 180												
296 - 335												
(N.News)												
119 - 124	Thorny	7000	265	732.5	.1046	48"	24"	174	1 3/8"	.120"	12	4 -
181 - 185	Croft					41/64 &	19/32 &	2482	1 1/8	.109		4 3/8"
						1 29/64"	7/8"					
(N.Y.Ship)												
125 - 130	Thorny	6762	260	727	.1075	50"	21"	176	1 3/8"	.120"	14	2 -
157 - 160	Croft					3/4" &	1/2" &	2304	1 1/8	.109		9"
						1 5/8"	29/32"					

FUEL OIL SYSTEM.

The system as installed on all of the vessels provides for the mechanical pressure atomization of the oil; there being either 12 or 14 atomizers and air registers fitted to each boiler.

In each fire room there is installed a booster pump, with connections for the transfer of oil from tank to tank and for the delivery of oil from any of the tanks to the service pump suction. These pumps also have connection to the deck-filling fitting and are arranged for pumping oil overboard, this connection being by means of a portable flanged pipe. On the Fore River and Union Iron Works built vessels, drainage suction pipes are led from the bottom of all oil tanks to a booster pump suction to permit pumping settled water from tanks and discharging it overboard.

There are two service pumps in each fire room, each pump being of sufficient capacity to furnish oil for both boilers in the compartment, when operating at full power. These pumps take suction through a strainer, either from the booster pump discharge or direct from the tank manifolds. They discharge through a finer mesh strainer, then through a fuel oil heater to the supply pipe on each boiler front. Independent connections, by means of flexible pipe, are taken from this supply line for each atomizer; there being a cutout valve in each connection, in addition to the operating valve at the atomizer on all except the Bath installations.

A hand pressure pump is provided for supplying oil to the atomizers, for raising steam with no air pressure available.

The size and make of fuel oil heaters and pumps as installed on the various groups of vessels, is in accordance with the following list:—

T.B.D. FUEL OIL HEATERS, PUMPS, ETC.

T.B.D. No.	F.O. Heaters No. and Type	H.S. each Heater Sq.ft	Coils or Tubes each Heater	Oil Conn. Ins.	Steam Conn. Ins.	Drain Ins.	Booster Pumps	Service Pumps	Make of Pumps
(Bath) 75 - 78	2-Vert. straight tube (oil in tubes -3 pass)	99.9	148-5/8" x .049" steel tubes	2	1	1	2-V.S. 4-1/2x14	4- V.S. 4-1/2x14-1/2x18	Warren
131 - 134									
244 - 247									
(N.Y.Ship) 125 - 130									
157 - 160									
251 - 250	"	"	"	"	"	"	2-V.S. 4-1/2x12	"	Worthington
(Maine Isl) 93 & 94	"	"	"	"	"	"	2-V.S. 4-1/2x12	"	Warren Pump
136 - 141									
234 - 241	"	"	"	"	"	"			
(Ch'ston) 135									
(N.Navy) 181 - 185	"	"	"	"	"	"	2-V.S. 4-1/2x12	4-V.S. 5-1/2x16	Worthington
186 - 199									
(Cramp) 115 - 118	2 Ver.return tube (oil in tubes)	80.0	87-3/4"x .065" steel tubes	2	1 1/4	3/4	2-V.S. 4-1/2x12 2-V.S. 4-1/2x12	4-V.S. 5-1/2x14-1/2x18	Warren Worthington
142 - 166									
236 - 239									
(Norfolk) 242 - 244									
(Fore R.) 79 - 84	2-Vert. Zimmerman37 straight tube oil in shell Water 1 pass	24.6	34-5/8"x .072"steel tubes	2	1	1	2 V.S. 4-1/2x12	4-V.S. 5-1/2x16	Worthington
93 - 102									
(U.Wks) 87 - 92									
102 - 112									
(N.Navy) 119 - 124	4-"Elite"								
(Fore R.) 141 - 170	2-Vert. Reilly 86 (oil in coils)	36.9	6 coils of 1"O.D. x .065" steel tubing	"	"	"	"	"	"
231 - 236									
(U.Wks) 171 - 180									
236 - 235									

The atomizers and air registers as installed on the various vessels all operate on the same principle and aside from Schutte and Koerting type of installations, the Bureau of Engineering standard atomizer tip and plug are in use.

All atomizers are so fitted as to be readily removed from the registers, overhauled or cleaned independently of each other. The following list gives the various types and number installed on the different groups of vessels:—

FORCED DRAFT BLOWERS.

Each vessel is provided with six high speed, turbine driven blowers, three in each fire room. They are all of the vertical, single inlet, semi-encased type, located close under the main deck and supported from foundations suspended from the deck beams and longitudinal girders. Short intake pipes connect the blowers with the individual trunks on the main deck; the discharge is directly into the fire room.

The blowers and turbines are self contained units of two different types, viz., the Sturtevant 42-inch Turbo vane, direct connected to a Sturtevant single stage, impulse turbine, and the Green Fuel Economizer 34-inch semi-silent fan, direct connected to a Terry single stage, impulse turbine.

Of the 90 vessels having Terry turbine installations, 36 have compounded units, there being one high pressure and two low pressure turbines in each fireroom. The steam connections are so arranged that any two units can be in operation with the third unit cut out, or either of the low pressure units can operate alone. The remaining Terry and all of the Sturtevant installations are high pressure units operating at 250 pounds steam and exhausting into the auxiliary exhaust line at about 10 pounds back pressure. Each turbine develops about 60 H.P. when operating under full power conditions.

T.B.D. FUEL OIL ATOMIZERS AND AIR REGISTERS

T.B.D. NO.	TYPE ATOMIZERS	TYPE REGISTER	AREA THRU REGISTER Sq.Ins.	NO. PER BOILER
(Bath) 75 - 78 131 - 134 345 - 347	Bath (tip practically like Bureau Engineering Standard.)	Bath (conical)	57.4	14
(Norfolk) 342 - 344	"	"	"	"
(Fore R.) 79 - 86 95 - 102 161 - 170 281 - 295	Bureau Engineering Standard.	F.R. (conical)	41.65	12
(U.I. Wks.) 87 - 92 103 - 112 171 - 180 296 - 335	"	"	"	"
(N. News) 119 - 124 181 - 185	"	Bureau Engineering natural forced draft.	111.0	12
(Mare Isl.) 93 & 94 136 - 141	"	"	"	"
(Cramp) 115 - 118 142 - 156	Schutte & Koerting.	Schutte & Koerting (conical).	77.0	14
(N.Y.Ship) 125 - 130 157 - 160	Bureau Engineering Standard.	Bu. Eng. type mod. to suit double front boiler.	93.5	14
(N. News) 186 - 205	Bureau Engineering Standard.	Bureau Engineering conical.	77.0	14
(Cramp) 206 - 230)	"	"	"	"
(N.Y.Ship) 231 - 250)	"	"	"	"
(Chs'ton) 135	"	"	"	"
(Mare Isl.) 336 - 341	"	"	"	"

T.B.D. FORCED DRAFT BLOWERS.

T.B.D. No.	Type Turbine	Fan	Capacity of each	R.P.M.	Steam Consumption Lb./Hr./	Ft.	Hrs.	Dia. Inlet	Width Outlet Ins./	No.
(Bath) 78 - 79 121 - 124 345 - 347	Terry G.V. (Compound)	Green Fuel Econ/Co.54" Semi silent	30000 cu.ft 8" air P.	1400				40"	7 5/8	2 HP 4 LP
(Cramp) 113 - 118 208 - 220	Terry G.V. (Simple)	"	30000 cu.ft 7" air P.	1470		265		"	"	6
(Herreshoff) 342 - 344	"	"	"	"						"
(NYShip) 125 - 130 157 - 160 231 - 250	Terry G.V. (Compound) (Simple)	"	30000 cu.ft 6" air P.	1425		250	4108			2 HP 4 LP 6
(Bare Isl) 93 & 94 136 - 141 336 - 341	Terry G.V. (Compound)	"	"	"		"	"	"	"	2 HP 4 LP
(Che'ton) 135	"	"	"	"						2 HP 4 LP
(Fore R.) 79 - 86 98 - 102 161 - 170 251 - 256	Sturtevant VD-5	Sturtevant 42" Turbo Vane	27000 cu.ft 7" air P.	1500	28/1000 cu.ft/ air/	265	4108	36"	6 1/2	6
(U.I. Wks) 87 - 92 103 - 112 171 - 180 236 - 235	"	"	"	"	"	"	"	"	"	"
(N. Penn) 119 - 124 161 - 165 186 - 199	"	"	30000 cu.ft 6" air P.	"	1.9#/HP cu.ft. air	"	"	"	"	"
(Cramp) 342 - 156	"	"	30000 cu.ft 7" air P.	1400	28/1000 cu.ft.air	"	"	"	"	"

FEED SYSTEM.

Both main air pumps, as well as the auxiliary condenser air pump, discharge into the filter compartment of the combined feed and filter tank, which is located in the forward engine room. This feed tank has a capacity of approximately 1100 gallons. A feed suction main is led from this tank to the suction of each of the feed pumps.

There are four feed pumps installed on each vessel; two main feed pumps in the forward engine room and an auxiliary feed pump in each fire room. They are all of the vertical, single cylinder, double acting, reciprocating type and each pump has a suction connection to the above mentioned suction main, as well as from the reserve feed tank.

The reserve feed tank has a capacity of from 24 to 25 tons and is located in the forward end of the forward engine room, being built into the hull of the vessel.

The main feed pumps discharge into a common line which extends forward into both fire rooms and makes connection with all boilers through feed stop and check valves. This discharge line makes connection to the feed water heaters on its way to the boilers, bypasses being provided, however, around each heater.

Each auxiliary feed pump discharges to both of the boilers in its compartment, through connections on the boilers which are entirely independent of the main feed connections. The auxiliary feed pumps also have connections to the inlet sides of the feed water heaters; when operating the auxiliary pumps on the heaters, however, the discharge to boilers is through the main feed connection.

There are two feed water heaters on each vessel; they are located in most cases one in each fire room; on the Bath, New York Ship and Mare Island installations, however, they are located in the forward engine room. The heaters are of various types; the requirement calling for their ability to

T.B.D. FEED WATER HEATERS, FEED PUMPS, ETC.

T.B.D. No.	F.W. Heaters No. and Type	H.S. each Heater Sq.ft.	Coils or Tubes: each Heater	Exh. Inlet Ins.	Drain Ins.	Feed Conn Ins.	Main Feed P. No. and Size	Aux. Feed P. No. and Size	Make of Pumps
(Bath) 78 - 79 121 - 124 240 & 242 (B.T.Ship) 125 - 120 137 & 140	2-Vert. straight tube	348.8	208-5/8" x .049" comp.A tubes	7 1/2	2	5	2-V.S. 16x12x24	2-V.S. 15x10x16	Warren
231 - 250	"	"	"	"	"	"	2-V.S. 14 1/2x12x24	"	Worthn
(Mare Isl) 92 & 94	"	"	"	"	"	"	2-V.S. 14x12x24	"	Warren
126 - 141 236 & 241 (Cr'atom) 136	"	"	"	"	"	"	"	"	Dow
(Cramp) 112 & 114 142 - 154 232 & 252 (Norfolk) 242 - 264	2-Hor. straight tube (long rad. bend to take expansion)	402	428-5/8" x.0625 comp.A tubes	8	2	5	" 2-V.S. 16 1/2x12x24	" 2-V.S. 14-1/2x12x18	Warren Worthn
(N.Navy) 119 - 124 181 - 185 186 - 199	2-247 Reilly	289	47 coils of 1"O.D x.065" copper t.	10	3	5	2-V.S. 16 1/2x12x18	2-V.S. 14-1/2x12x18	Worthn
(Fore R.) 79 - 86 91 - 102 (U.I.Wks) 87 - 92 103 - 112	2-Hor. S. & R. straight tube	448	282-5/8" x.065" comp.A tubes	7	3	4	"	"	"
(Fore R.) 161 - 170 251 - 272 (U.I.Wks) 171 - 180 296 - 335	2-439 Reilly	240	39 coils of 1"O.D x.065" copper tube	"	"	"	"	"	"

heat all feed water required at full power from 90 degrees F. to 225 degrees F. with 240 degrees F. temperature of the heating steam. This latter temperature corresponds to about 10 pounds steam as furnished by the auxiliary exhaust line, on which the feed water heaters operate.

AIR COMPRESSOR PLANT.

One 11-inch × 11-inch × 12-inch Westinghouse, vertical, steam driven, double acting air compressor, is installed in the after fire room, to supply air for the charging of air chambers, for the cleaning of boilers, generators, fuel oil atomizers, etc., and also for various purposes when steam is not available.

The compressor is a single stage unit, usually operated on these vessels at a pressure of about 100 pounds, and is connected with an air reservoir of about 20 cubic feet capacity, which is installed close by. Each compressor is fitted with a governor which automatically maintains an air pressure in this storage tank, at the point for which it is set. The governor is entirely automatic after being adjusted for the pressure desired.

While some of the compressor air cylinders are provided with water cooling jackets, many are simply air cooled and operate satisfactorily in service.

A second compressor of the same type and size as the original has been installed on vessels fitted with 5-inch guns, for gas ejecting purposes.

EVAPORATING AND DISTILLING PLANT.

The requirement for the earlier vessels of this program was for an evaporating plant of 7500 gallons capacity per 24 hours; but this was subsequently increased, and beginning with the 161-170 group of vessels, 10,000 gallon capacity plants have been installed. These ratings are for two evaporators operating in double effect, with the usual Navy test re-

quirement for an overload capacity, when tubes or coils are clean, of at least 40 per cent in excess of these nominal capacities; the distilled water to contain not more than 2 grains of chlorine per gallon.

The evaporating plant complete is located in the after engine room on all these destroyers except those built at the Navy Yards, Mare Island and Charleston, in which vessels a third evaporator with its distiller and pumps, was installed in the after fire room.

The plants all consist of two, or of three in some of the later groups, vertical distillers having the full capacity of their respective evaporators, a 4½-inch × 6-inch × 6-inch vertical simplex evaporator feed pump, a feed water heater, a 3¼-inch × 4-inch × 4-inch V.S. distiller fresh water pump, and two 50 gallon distiller test tanks. Circulating water for the distillers is furnished by the 7-inch × 7-inch × 12-inch V.S. fire and bilge pump located in the after engine room.

The plants are all piped up for operation in either single or double effect, steam being furnished at 70 pounds gauge. The distilled water is discharged by the fresh water pump to a line connecting with the main feed tank, reserve feed tank, ships fresh water tanks, gravity tank and to the cofferdam.

The evaporators are mostly of the Reilly submerged coil type, with heating surface and other particulars as given in the following list.

Continuous efforts are being made to obtain the highest possible efficiency of the evaporating plant on these vessels, due regard being given to the necessarily light and compact character of the plant on vessels of this type. Reports indicate a growing tendency to connect up the evaporators with the auxiliary exhaust line, and operate as much as possible on exhaust steam; this condition is particularly advantageous when operating in port. The carrying out of the next step toward greater economy has been effected on a number of the

T.S.D. EVAPORATORS.

T.S.D. NO.	Spec. Cap. Plant (D.Eff)	No. & Type	H.S. each Evap.	SHELL Feed Ins.	SHELL Vapor Ins.	SHELL Flow Ins.	COILS (or Tubes) Steam	COILS (or Tubes) Drain	Mat'l Shell
(Bath) 75 - 78 151 - 154	7500	2 @16 Reilly	42.56	1	3 - 1st Eff 3 1/2-2nd Eff	2	2-1st Eff 3-2nd Eff	1 1/2	steel plate
(Cramp) 113 - 118 142 - 146	"	"	"	"	"	"	"	"	"
(N.Y.Ship) 125 - 130 137 - 140	"	"	"	"	"	"	"	"	"
(Mare Isl) 93 & 94	9500	* 3 @16 Reilly	"	"	3 - 1st Eff 3 1/2 2nd Eff	"	"	"	"
(N. News) 119 - 124	7500	2 @16 Reilly	"	"	3 1/2	2 1/2	2 1/2	2	Comp. C.
(Fore R.) 79 - 86 95 - 102	"	"	"	"	"	"	"	"	"
(U.J. Wks) 87 - 92 103 - 112	"	"	"	"	"	"	"	"	"
(Fore R.) 141 - 170 251 - 292	10,500	2 @24 Reilly	65.84	1 1/4	4	2 1/2	3 1/2	1 1/2	"
(U.I. Wks) 171 - 180 226 - 235	"	"	"	"	"	"	"	"	"
(N. News) 181 - 192	"	"	"	"	"	"	"	"	"
(Cramp) 206 - 220	"	"	"	"	"	"	"	"	"
(N.Y.Ship) 221 - 250	"	"	"	"	"	"	"	"	"
(Norfolk) 242 - 244	"	"	"	"	"	"	"	"	"
(Mare Isl) 136 - 141 236 - 241	9000	* 3 Ru.Std. Eng.	50/75	1	3 1/2	"	2 1/2	2	"
(Ch'ston) 135	"	"	"	"	"	"	"	"	"
(Bath) 245 - 247	10,000	2 - U Tube (Bath)	110	"	3	3	2	1	"

* Subject to test at 40% over spec/ cap.
* One in Aft Engin., one in Aft Fire R.

T.B.D. DISTILLERS.

T.B.D. No.	Spec. Cap. Plant (G.P.G.)	No. & Type	G.S. each Dist. Sq.ft	Coils or Tubes each Distiller	Vapor Inlet Ins.	Vapor Outlet Ins.	Circ. Water Inlet Ins.	Circ. Water Outlet Ins.	Mat'l Shell
(Bath) 75 - 76 151 = 154 (Cramp) 115 - 118 142 = 146 (N.Y.Ship) 125 - 130 147 = 150 (Mare Isl) 93 & 94	7500	2-Vert. straight tube (vapor in tubes)	63.49	138-5/8" O.D.x.049" comp.tubes	2 1/2	1 1/4	2	2	Seamless drawn brass tube .109" thick
136 - 141 154 = 241 (Ch'ston) 135	9500	2-Vert. straight tube (vapor in coils)	"	"	"	"	"	"	"
(Bath) 242 = 247 (Cramp) 236 = 240 (Norfolk) 242 - 244	10,000 10,500								
(N.Navy) 119 - 124 (Pure R.) 78 - 84 96 = 102 (U.I.Nav) 87 - 92 108 - 112	7500	2-36 Reilly	49.2	8 coils of 1"O.D.x.065 copper tube	3	1 1/4	2 1/2	2 1/2	cast iron 1/2"thick
(N.Navy) 186 = 192 (Pure R.) 161 - 170 251 = 275 (U.I.Nav) 171 - 180 226 - 235	10,500	2-315 Reilly	79.9	13 coils of 1" O.D. x .065" copper tube	3 1/2	3	2 1/2	2 1/2	cast iron 1/2"thick
(N.Y.Ship) 221 - 230	10,500	2-Vert. straight tube	95.0	208-5/8" O.D.x.049" tubes.	3 1/2	1 1/2	3	3	copper .125"thick

* Subject to test at 40% over spec. cap.
Note - One in Aft Eng.R., one in Aft Fire R.for Mare Island and Charleston T.B.D.'s.

vessels by connecting up certain pumps in the after engine room, one for use as a distiller air pump and another as a brine pump. These developments may result in providing proper equipment and connections for normally operating these plants as low pressure systems, particularly as considerable exhaust steam is available even under ordinary cruising speeds.

FIRE AND BILGE PUMPS.

There is a 7-inch × 7-inch × 12-inch vertical simplex, double acting fire and bilge pump located in each engine room and in each fire room, each pump having suction from the drainage system and from the sea and discharge connections to the fire main and overboard. The pumps in the engine rooms are connected to the flushing system through the fire-main and the pump in the after engine room also supplies circulating water to the distillers.

REFRIGERATING PLANT.

Each vessel is equipped with a large refrigerator box equipped with cooling coils; it is located on and firmly secured to the main deck, in the vicinity of the galley.

The refrigerating machine is installed in combination with the box and except in some of the earlier vessels, is located directly on the box. The various types of machines are as given in the following list:—

MACHINE SHOP.

The machine shop, which is located in the forward fire room, has the following tools:—

1 motor driven screw-cutting back-geared engine lathe with a 12-inch swing and a 6-foot bed.

1 motor driven double emory grinder.

1 motor driven 24-inch two-speed Universal bench drill.

T.B.D. REFRIGERATING APPARATUS.

T.B.D. No.	No.	Type	Capacity	Refrigerant used	Type of Cooling System	R.P.M. of Machine	Driving Unit Size	Driving Unit R.P.M.
(Bath) 75 - 78 131 - 134 (Cramp) 115 - 118 (Mare Isl) 93 & 94 126 - 141 (Cramp) 135 (Fore R.) 79 - 86 95 - 102 161 - 170 (U.E.Wks) 87 - 92 103 - 112 171 - 180	1	86 Audiffren Singrun (Johns - Manville Co)	1/2 ton	Sulphur Dioxide	Brine Circulation	280	1 1/2 HP Motor	1150
(Cramp) 142 - 156 206 - 220 (Mare Isl) 226 - 241 (N.Y.Ship) 125 - 130 157 - 160 (N.Navy) 119 - 124 181 - 186 196 - 199 (Norfolk) 242 - 244	1	Clothel	1/2 ton	Ethel Chloride	Direct Expansion	900	1 1/2 HP Motor	900
(N.Y.Ship) 221 - 250	1	York	1/2 ton	Carbon Dioxide	Direct Expansion	500	2 H.P. Motor	800
(Bath) 245 - 247 (Fore R.) 251 - 256 (U.E.Wks) 296 - 325	1	82 Audiffren Singrun (Johns - Manville Co)	1/5 ton	Sulphur Dioxide	Brine Circulation	280	3/4 HP Motor	1145

In addition to the above, there are provided the usual hand tools and one blacksmith's forge.

ELECTRIC PLANT.

This plant consists of two 25 K.W. turbine driven generating sets with their switch board, etc., located on a platform in the after engine room. These units are of either the General Electric Company or Westinghouse make and operate on 125 volts, D. C.

TRIALS.

At least one vessel of each group had a complete set of standardization trials, as well as four hour runs at various speeds, including one at full power.

Tabulated below are the data obtained from the trials of all vessels having the full set of trials; included therein, it will be noted, are the capacities of fuel oil and reserve feed tanks, machinery weights for the various types of installations as well as shaft horsepower and fuel consumption at the several speeds.

T.B.D.'s PRELIMINARY TRIAL DATA.

Name of Ship Where Built	Manley (74) Bath	Conner (72) Cramp	Wickes (75) Bath	Bell (95) Fore R.	Palmer (161) Fore R.	Robinson (68) U.I.Wks	Champlin (106) U.I.Wks	Burns (171) U.I.Wks	Howard (179) U.I.Wks	Gamble (123) N.Navy		Dent (116) Cramp	Tarbell (142) Cramp
Av.Speed K.	32.23	30.126	35.34	34.75	35.60	34.28	33.54	33.82	34.75	35.01		35.11	35.12
Av. S.H.P.	19715	20060	24610	27240	29165	28425	28194	28900	29675	24520		26025	24675
Total oil per hr.lbs.	19175	20755	22432	25554	30862	29064	30410	28111	28522	27973		28946	23696
# Oil/SHP hr.	.975	1.034	.912	.975	1.089	1.023	1.079	.9727	1.089	1.141		.894	.912
" " "sq.ft. H.S.	.692	.922	.83	.965		1.054	1.130			.998		.828	
" " "Knot	595@32½ K.	689@30.1 K.	634.75 @ 35.14 K.	774@35K	811.8 @35 K.	850@34½ K.	906.68 @33.54K	851@33.8 K.	935@34K K.	847.39 @33.01K		648@35 K.	641@35 K.
" " " at 30 K.			454	558	515	600	625	596	671	735		484	462
" " " " 25 K.			261	347	346	422	382	395	400	474		280	317
" " " " 20 K.			163	202	196	245	244	264	258	250		180	204
" " " " 15 K.			121	155	145	165	195	175	192	190		130	140
Cap.F.O.Tanks (Tons)			288 ⊘	274 *	274 *	273.35 *	275 *	272 *	272 *	274 *		288	288
Cap.H.F.Tanks (Tons)			44.83	25.25	25.25	22.50	25.32	25.51	25.51	24.26		24.3	24.3
Wt.Machy (wet) (tons)			421.94	465.69	467.81	476.98	486.46	488.07	482.94	467.4		423.65	429.02
Wt. Water (Tons)			43.70	46.56	47.10	49.05	48.69	49.15	45.07	39.40			
Av.Disp.Tons			1149	1188	1187	1247	1214	1230	1231	1206		1164	1162
S.H.P.at 35K.			23975		27500	30500						25675	24560
" " " " 30"			14500	17600	16900	18000	17900	18175	19300	17250		16200	14850
" " " " 25"			7300	8760	8740	9000	8600	9050	9550	9120		8400	7835
" " " " 20"			3725	3510	3400	3750	3500	3450	4040	3740		3300	3300
" " " " 15"			920	1440	1420	1050	1040	1275	1400	1110		1250	1240
Water.Total	245,011	294,101		378,166									
" " M.Eng.	223,488			324,997									
" " Aux.	21,623			53,169									
Water/SHP Total	12.43	14.66		13.88									
" " " Main Engll.	11.31			11.95									
" " " Aux.	1.12			1.95									

* 95% full

⊘ including cofferdam

Dent (116) Cramp	Tarbell (142) Cramp	Boggs (136) Mare I.	Kennison (139) Mare I.	Jacob Jones (130) N.Y.Ship	McDermut (262) Fore R.	Belknap (251) Fore R.	Farragut (300) U.I.Wks	Nicholas (311) U.I.Wks	Chandler (206) Cramp	Brooks (232) N.YShip		Satterlee (190) N.Navy	
5.11	35.12	35.77	35.89	35.08	34.72	34.63	32.68	33.74	34.75	33.25		34.99	
025	24875	27250	28500	25320	28350	28728	27750	28600	25160	27610		25480	
944	23886	27610	29855	25044	27599	30572	28584	31188	23857	26948		27245	
24	.912	1.014	1.055	.990	.976	1.064	1.023	1.091	.948	.976		1.029	
28													
@35 K.	641@35 K.	722@35 K.	852@35 K.	714@35.8 K.	794@34.7 K.	909@35 K.	870@ 34.9 K.	950 @ 34.74 K.	686 @ 34.7 K.	810 @ 33.26 K.		779@35 K.	66
4	462	470	490	482	589	585	650	650	495	618		564	48
0	317	342	350	318	366	352	400	440	296	360		384	34
0	204	240	202	208	206	205	297	272	198	222		232	18
0	140	156	156	148	158	149	257	206	154	202		176	15
8	288	274 *	274 *	274 *	371	371	375	375	370	376		394.2	37
.5	24.5	41.08 @	41.08 @	24.23	25.25	25.25	21.	21.	25.50	22.57		24.10	44
5.65	429.02	453.40	459.85	439.81	469.21	472.47	485.	486.61	451.56	472.25		474	43
		42.57	42.57		46.65	46.64	43.52	43.52		35.72		38	4
64	1162	1200	1180	1166	1225	1257	1292	1291	1225	1305		1282	12
375	26560	25975	26350	25120	29500	29650	36000	34000	25660	32600		26500	27
300	14850	16760	15500	15300	18600	19400	20500	21500	16300	20300		16800	17
400	7655	8460	7650	7800	9250	9250	10250	10800	7925	9450		8900	8
300	3300	3990	2250	2950	3800	3780	4300	4000	3120	3620		3700	3
250	1840	1230	700	910	1440	1560	1700	1750	1230	1380		1400	1
								304,710					34 31 2
							11.35	10.91					12 11 1

...R.B.D.'s PRELIMINARY TRIAL DATA.

...r	Wickes (75) Bath	Bell (95) Fore R.	Palmer (161) Fore R.	Robinson (88) U.I.Wks	Champlin (104) U.I.Wks	Burns (171) U.I.Wks	Howard (179) U.I.Wks	Gamble (123) N.Yeara		Dent (116) Cramp	Tarbell (142) Cramp	Boggs (136) Mare I.	Kennison (189) Mare I.	Jacob Jones (150) N.Y.Ship	McDermut (262) Fore R.
...6	35.54	34.75	35.60	34.29	33.54	33.82	34.75	33.01		35.11	35.12	35.77	35.89	35.08	34.72
...0	24610	27240	29165	28425	28194	28900	29875	24520		26025	24875	27250	28500	25520	28350
...5	22452	26554	30862	29064	30410	28111	32522	27973		23946	21998	27610	29855	25064	27599
...4	.912	.975	1.089	1.023	1.079	.9727	1.089	1.141		.896	.912	1.014	1.055	.990	.976
...2	.83	.965		1.054	1.130			.998		.828					
...0.1 K.	654.75 @ 35.34 K.	774@35K	811.8 @35 K.	850@34.2 K.	906.68 @ 33.54K	851@33.8 K.	985@34.7 K.	847.39 @33.01K		645@35 K.	641@35 K.	722@35 K.	852@35 K.	714@35.8 K.	794@34.7 K.
	434	556	515	600	625	596	671	735		484	462	470	490	482	589
	261	347	346	422	382	395	400	474		280	317	342	350	318	366
	183	202	196	245	244	264	256	250		190	204	240	202	203	206
	151	155	143	165	193	175	192	190		130	140	156	156	143	158
	288	274 *	274 *	273.35 *	273 *	272 *	272 *	274 *		288	288	274 *	274 *	274 *	371
	44.85 (2)	25.25	25.25	22.50	25.32	25.51	25.51	24.26		24.5	24.5	41.05 (2)	41.05 (2)	24.23	25.25
	421.94	465.69	467.81	476.98	486.46	488.07	482.94	467.4		423.65	429.02	455.40	459.85	439.81	469.21
	43.70	46.56	47.10	49.05	48.69	49.15	45.07	39.40				42.57	42.57		46.65
	1149	1188	1187	1247	1214	1230	1231	1206		1164	1162	1200	1180	1166	1225
	28975		27500	30500						25875	24560	25975	26350	25120	29500
	14500	17600	16900	18000	17900	18175	19300	17250		16200	14850	16760	15800	15800	18600
	7500	8760	8740	9000	8600	9050	9650	9120		8400	7835	8460	7650	7800	9250
	3725	3510	3400	3750	3500	3450	4040	3740		3300	3200	3390	2250	2950	3560
	980	1440	1480	1050	1040	1275	1400	1110		1250	1240	1230	700	910	1440
...01		578,166 324,997 55,169													
...66		15.88 11.93 1.95													

* 95% full
(2) including cofferdam

Jones (0) Ship	McDermut (262) Fore R.	Belknap (251) Fore R.	Farragut (300) U.I.Wks	Nicholas (311) U.I.Wks	Chandler (206) Cramp	Brooks (232) N.Ship		Satterlee (190) N.News	Pruitt (347) Bath	Wadsworth (344) Norfolk	Litchfield (336) Mare I.
08	34.72	34.63	32.68	32.74	34.75	33.26		34.99	35.11	35.14	34.92
20	28350	28720	27750	28600	25160	27610		28480	27875	22800	29900
44	27599	30572	28384	31188	23857	26948		27245	23455	25992	32071
0	.976	1.064	1.023	1.091	.948	.976		1.029	.841	1.184	1.071
35.08 K.	794@34.7 K.	909@35 K.	870@ 34.43 K.	950@ 34.74 K.	686@ 34.75 K.	810@ 34.26 K.		779@35 K.	668@ 35.11 K.	768@35.14 K.	921@35 K.
	589	585	650	680	485	618		564	459	563	700
	356	352	400	440	296	360		384	348	384	504
	206	205	297	272	198	222		232	185	264	339
	158	149	257	206	154	202		176	133	198	253
*	371	371	375	375	370	376		394.2	375	370	271
3	25.25	25.25	21.	21.	25.50	22.97		24.10	44.85	25.50	41.09
11	469.21	472.47	485.	486.61	451.86	472.26		474	455.31		469.52
	46.65	46.64	43.52	43.52		35.73		38	41.65		42.52
	1225	1257	1292	1291	1225	1303		1382	1214	1225	1285
)	29800	29650	36000	34000	25640	32600		26500	27600	22650	30200
)	18600	19400	20800	21300	16300	20200		16800	17040	15100	17450
)	9250	9280	10250	10800	7925	9450		8800	8470	7800	8850
)	3680	3780	4200	4000	3120	3620		3700	3420	3100	3600
)	1440	1360	1700	1750	1230	1350		1400	1340	1340	1525
				304,710					342,716 313,383 29,333		409,184
			11.25	10.91					12.295 11.293 1.002		13.685

SEAGOING TUGS, NOS. 21 TO 39, INCLUSIVE.

DESCRIPTION AND TRIAL DATA.

By Lieut. Comdr. H. G. Donald, U. S. Navy, Member.

GENERAL DESCRIPTION.

As a part of the War building program there were added to the Navy nineteen exceptionally powerful Seagoing Tugs. A description of these vessels should prove of interest to the service at large, not only because of their recent construction, but also because of their assignment to the Fleet as tenders and to the Navy Yards for heavy duty work. In size and power they rate as high as any Seagoing Tugs in the United States and in general equipment are superior to any in commercial service.

A list of the vessels follows, giving Navy Classification Number, Name, Builder, Date of Laying Keel, Date of Launching and Date of Acceptance.

TABLE I

Navy No.	Name	Builder	Keel laid	Launched	Accepted
1021	MOHAWK	Marvine Steel & Iron Works, Buffalo, N.Y.	July 10,1918	April 5,1919	Sept.10,1919
1022	TAWASA	" " " " " "	July 10,1918	Nov. 15,1919	Sept. 15,1919
1023	KIOWA	" " " " " "	Sept. 20,1918	Nov. 15,1919	Nov. 14,1919
1024	LOGANIA	" " " " " "	Sept. 20,1918	Dec. 20,1919	Nov. 4,1919
1025	CHOCTA	" " " " " "	Sept. 20,1918	April 10,1919	Dec. 15,1919
1026	SEMINOLE	" " " " " "	April 1,1919	June 15,1919	Dec. 15,1919
1027	TUSCORA	Navy Yard, Puget Sound, Wash.	Nov. 5,1918	Nov. 11,1919	Jan. 30,1920
1028	APACHE	" " " " "	Nov. 5,1918	Nov. 11,1919	Jan. 30,1920
1029	ACCOMAC	" " " " "	Nov. 20,1918	Dec. 20,1919	Feb. 28,1920
1030	SECOTAN	" " " " "	Nov. 20,1918	Dec. 20,1919	Feb. 28,1920
1031	ISOLA	" " " " "	Nov. 20,1918	Jan. 15,1920	Mar. 15,1920
1032	NAHA	" " " " "	Nov. 20,1918	Jan. 15,1920	Mar. 15,1920
1033	PEMAQUID	" " " " "	Dec. 5,1918	Feb. 15,1920	May 1,1920
1034	ALGONAC	Staten Is. Shipbuilding Co., Shooters Island, Staten Is., N.Y.	Dec. 5,1918	Feb. 15,1920	May 15,1920
1035	CHICKAHOMINY	" " " " " "	Dec. 5,1918	Feb. 15,1920	June 30,1920
1036	SAUTUCKET	" " " " " "	Dec. 5,1918	Mar. 15,1920	June 30,1920
1037	ESKI	" " " " " "	Dec. 5,1918	Mar. 15,1920	July 30,1920
1038	WALLAMA	" " " " " "	Dec. 5,1918	Mar. 15,1920	Aug. 30,1920
1039	SUNDIAN	" " " " " "	Dec. 5,1918	Mar. 15,1920	Sept. 30,1920

All of these vessels were constructed under authority of the Act of Congress, dated March 4, 1917, which act made appro-

priation for the Naval service for the fiscal year ending June 30, 1918, the appropriation for these tugs being provided by the subhead "Naval Emergency Fund." Tugs No. 21 to No. 26, inclusive, were contracted for at a fixed price of $550,000 per vessel. Tugs No. 27 to No. 33, inclusive, were built by the Government at a cost of $474,370 per vessel. Tugs No. 34 to No. 39, inclusive, were built under contract providing for fixed profit of $37,500 per vessel providing the actual cost exceeded $500,000, or $37,500 plus one-third of the saving in cost below $500,000. The cost of each vessel of this group was $554,948. The original contract for this last group called for tugs No. 34 to No. 45, inclusive, but in December, 1918, the construction of tugs No. 40 to No. 45, inclusive, was cancelled.

The trials provided for by the contract were as follows:

(*a*) Three runs at full speed over a measured mile course.

(*b*) Six-hour full speed trial, on which it was expected to attain 13 knots speed and a fuel consumption not in excess of 200 pounds of fuel oil per knot run.

(*c*) Turning and backing trials.

The first vessel completed in each group was required to have all of the above trials. The remaining vessels were subject only to such trials as were considered necessary by the trial board to demonstrate that the vessel in question was capable of fulfilling the same tests and requirements as the first vessel of the group.

In the No. 21 to No. 26 group, the *Bagaduce* (21) was designated for full trials, although this vessel was the second one of the group to be completed. July 29, 1919, the *Bagaduce* was given a six-hour full power trial in Lake Erie by the Board of Inspection and Survey. On this trial, the expected speed of 13 knots and horsepower of 1,800 were not attained but the vessel was conditionally accepted. On April 28, 1920, a second full power trial was held by the Board of Inspection and Survey, but the speed and indicated horsepower developed were

still much below expected results. The Board reported that it believed this failure to develop the expected speed was due to an error in the propeller. It was learned at a later date that the propeller installed on this group of tugs was of such dimensions that the rated horsepower could not be developed. The propellers are being replaced with others of correct dimensions as opportunity offers, and the *Wandank* was given a standardization and full power trial on May 26, 1921, with the new propeller. Data taken on the trial appears under Table IV. The performance on this trial was little better than the second trial of the *Bagaduce*. The failure to develop greater power is believed to be due to the generally poor character of workmanship in both machinery and hull.

In the No. 27 to No. 33 group, the *Tatnuck* (27) was given a full set of trials under the supervision of a board appointed by the Commandant of the Puget Sound Navy Yard. In addition, a four-hour run was made at 10 knots under natural draft. These trials were held off Vashon Island on January 19, 20 and 21, 1920. Data are given in Table IV.

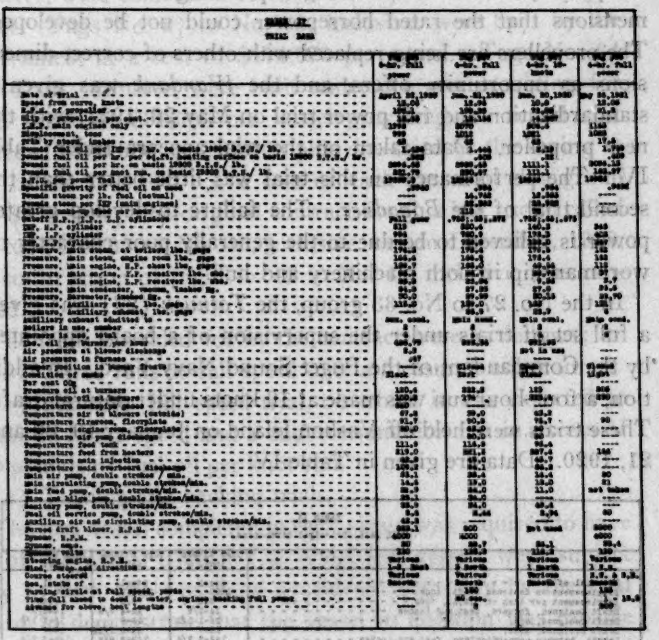
TABLE IV

FULL DATA. Principal Dimensions.

Material of hull	Steel	Steel	Steel
Displacement on designed draft, tons	1000	1011.89	1000
Displacement per inch on L.W.L., tons	9.6	9.6	9.58
Draft designed, forward, feet and inches	14-1/2	13-7 7/8	14-1 7/8
Draft designed, aft,	15-1 1/2	15-3	15-1 1/2
Draft designed, mean	14-7	14-5 7/8	14-7
Length, between perpendiculars, feet and inches	140-6 3/4	140-6 3/4	140-6 3/4
Length, over all, feet and inches	154-6	154-6 3/4	154-6
Length, on L.W.L.,	144-6 3/4	144-6	144-6
Beam, extreme,	31-6	31-6	31-6
Beam, on L.W.L.,	29-6	29-6	29-6
Beam moulded,	29-0	29-0	29-0
At designed draft, area immersed midship section, sq.ft.	340	338	343
At designed draft, area L.W.L. plane, sq.ft.	2880	2881	2880
At designed draft, wetted surface, sq.ft.	4100	4100	4098
Coefficient of fineness, block	.666	.66	.66
Coefficient of fineness, midship section	.809	.788	.796
Coefficient of fineness, L.W.L.	.704	.704	.702
Speed, designed, knots	14.0	13.9	14.0
Capacity bunkers fuel tanks, gallons	47.30	4870	4810
Capacity, ship's fresh water tanks gals. (including peak tanks)	9760	9760	9760
Capacity fuel oil tanks gals. @ 300 gals./ton	90,000	90,000	90,000
Cruising radius at full speed, nautical miles	5000	5000	5000
Metacentric height, full load 2046 tons displ. (no part of ammunition) draft 20r'4 12-8 1/2, aft 14-8 3/4, mean 14-10 5/8,			
fuel oil 60,000 gallons, feet		2.47	
Cruising radius at 60,000 gal. oil @ 10 knots, knots	5100	5100	5100
Cruising radius on 60,000 gal. oil @ 11 1/2 knots, knots	5400	5400	5400
Ballast placed in ptks., fuel oil tank to correct list to port, tons		15	

In the No. 34 to No. 39 group, the *Algorma* (34) was given full power and backing trials by the Board of Inspection and Survey on April 20, 21 and 22, 1920, in Long Island Sound.

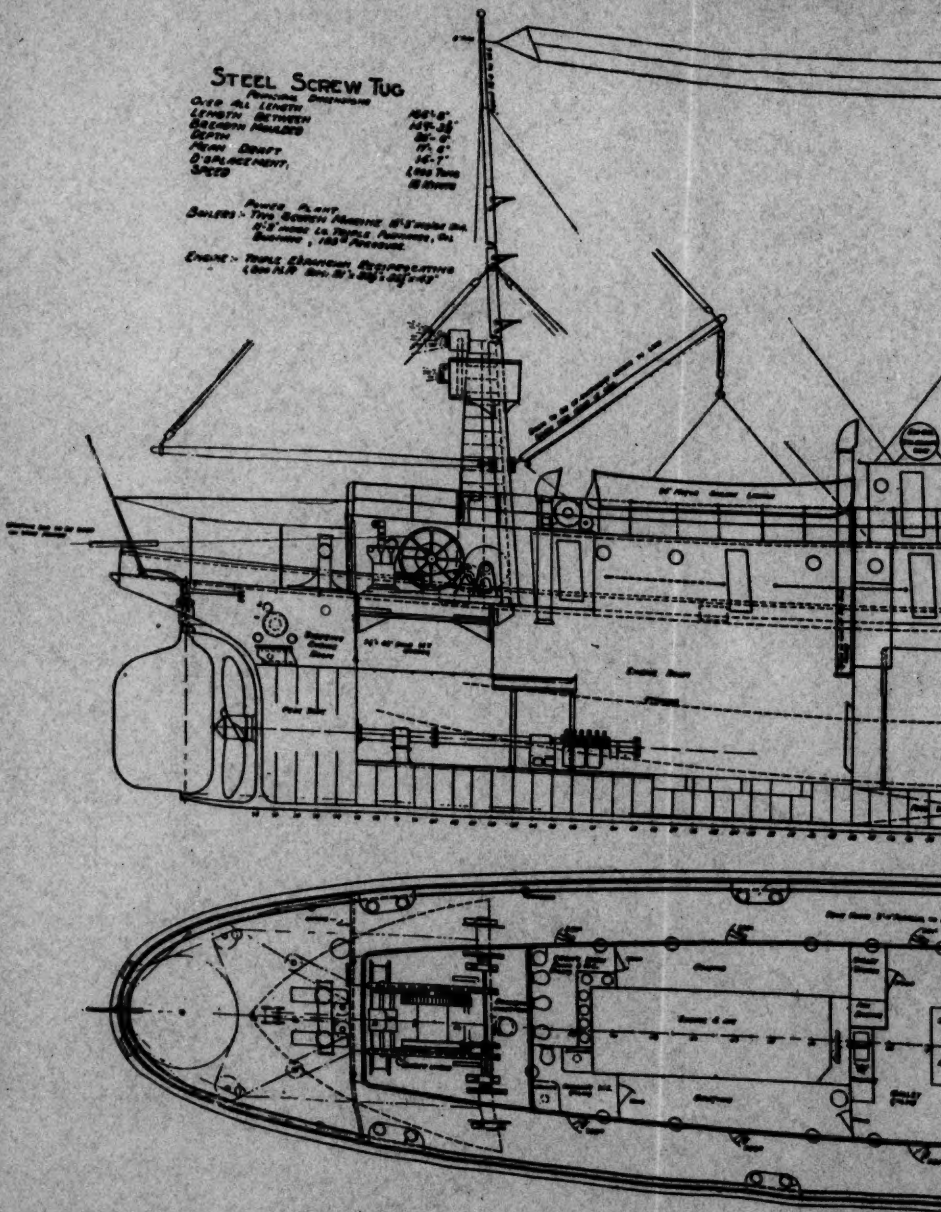
The standardization trial was held July 21, 1920, on the Provincetown, Mass., measured mile course. Data obtained on these trials are given in Table IV.



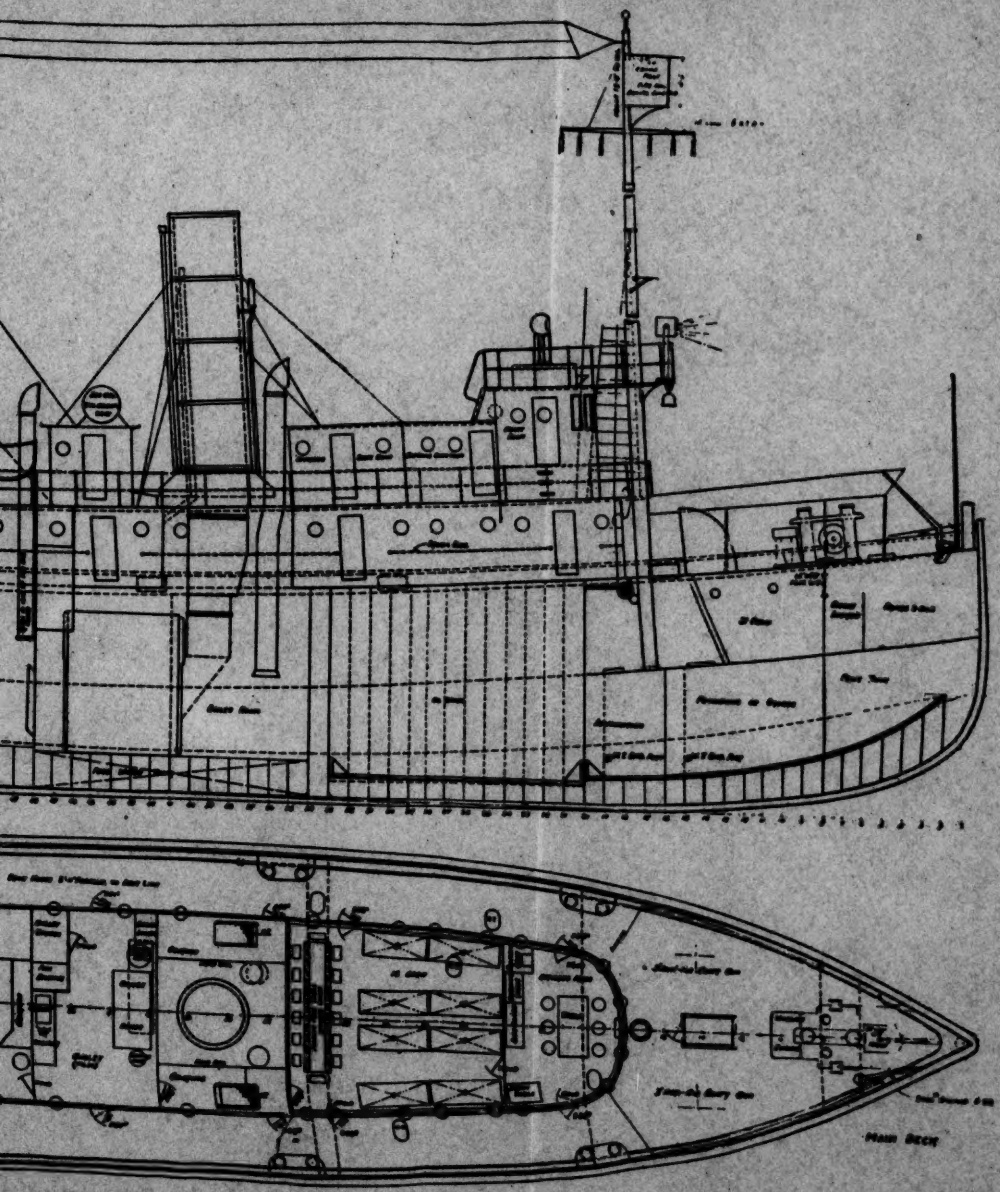
DESCRIPTION OF HULL.

These vessels are single-deck, single-screw, steam tugs, built of steel and intended for sea towing and mine sweeping. The general appearance is as shown on plate 1 and the photograph.

The transverse system of framing is used, with a frame space of 22 inches. The stern is straight above the waterline and the forefoot is cut away as shown. The stern is of the fantail type. Above the main deck there is a steel deckhouse, enclosing crew's berthing accommodations, galley stores, galley, officers' and crew's toilets, engine and boiler casings and offi-



SEAGOING TUGS



GOING TUGS NOS. 21–39.

cers' mess. On top of the deckhouse, inclosures of steel and brass construction provide for chart house, officers' staterooms and radio room. Below the main deck there are steel platforms, one forward and one aft.

There are two steel pole masts, with crow's nest fitted on the foremast. The main mast carries two booms for the handling of boats and weights over the stern, each boom being tested to a working load of 6,726 pounds.

Fuel oil is carried in athwartship compartments between frames 20 and 34 and extending from the keel to the main deck. The normal capacity is listed in Table II. Due to low freeboard and lack of stability the fuel oil capacity of tugs No. 19 to No. 39, inclusive, has been limited to 60,000 gallons, except in case of emergency requiring full tanks.

Reserve feed water is carried in a double bottom under the boiler room. The capacity is as listed in Table II.

A bulwark, two feet high, measured above the moulded deck line, runs the entire length of the vessel, increasing to a height of three feet at the bow. In the bulwarks there are freeing ports having a total area of 5 per cent of the bulwark area.

A bilge keel extends from frames 21 to 56 on either side of the vessel.

Transverse watertight bulkheads are located at frames 8, 20, 33, 34, 48 and 73, extending from keel to main deck; and at frames 16 and 19 from keel to platform deck only.

Provision is made for stowing the following boats on the after part of the superstructure deck:

One 24-foot motor sailing launch.
One 24-foot whale boat.
One 16-foot dinghy.

The battery was to consist of two 3-inch 50-caliber anti-aircraft guns and one machine gun. The 3-inch guns have not been provided although the foundations are installed. In the magazine in the fore hold, there is stowage space for 600 rounds

of 3-inch ammunition and 1,200 rounds of 30-caliber machine gun ammunition besides a considerable amount of rifle and revolver ammunition.

Berthing and messing arrangements are provided for a complement of 5 officers, 2 chief petty officers and 45 enlisted men of lower ratings.

DECK MACHINERY.

For handling the anchors and chains there is a steam windlass, Hyde type, engine (2) 6 inches × 6 inches stroke, capable of hoisting and lowering one bower anchor at the rate of 6 fathoms per minute.

A steam steering engine, Lidgerwood type, engine (2) 5 inches × 6 inches stroke, is located on the platform deck aft. The engine can put the rudder from hard-over to hard-over in 20 seconds with the vessel moving at full speed ahead. The engine may be controlled from either a steering wheel in the chart house or an emergency steering stand located aft on the superstructure deck.

A double drum, double gypsy, Lidgerwood, steam winch, engine (2) 8¼ inches × 10 inches stroke, is installed on the superstructure deck abaft the mainmast for handling the two booms on this mast, and also the boats. The winch is capable of handling, 6,720 pounds at 120 feet per minute.

A steam towing machine, built by the American Engineering Company, is located on the main deck aft. It has a capacity for a 2-inch diameter steel wire hawser, 300 fathoms in length. The engine is two-cylinder simple, 16 inches × 16 inches stroke.

The galley has one 3-foot section oil range, one steam-jacketed 20-gallon copper kettle, one coffee urn and one motor-driven food grinder.

All ventilation of spaces below deck is natural, except the crew's quarters forward below the main deck which are artificially ventilated by a motor-driven fan of 1,000 cubic feet per minute capacity.

Heating is provided for the living quarters only.

PIPING SYSTEMS.

The fire system consists of a six-inch main under the main deck extending between frames 16 to 73, with risers from the fire and bilge pumps, and branch connections to 2½ inches fire plugs, one forward, one aft, and one on each side of the deck house amidships, and a 4-inch monitor nozzle on top of the pilot house.

A salt-water sanitary system supplied by the sanitary pump, provides water for water closets, urinals, sink and faucets in crew's wash room and galley. A by-pass is installed from the fire main for emergency use.

The fresh water system is supplied from a 300-gallon gravity tank on top of the radio house, with branch supply pipes to the galley, lavatory and shower in officers' toilet, lavatories in officers' staterooms, lavatories and shower in crew's toilet and a faucet on deck for filling buckets. Water is pumped from the fresh water compartment forward to the gravity tank by a fresh water pump located in the crew's quarters forward.

The drainage system consists of a drain line running from the forward peak tank to the after tank, with independent suctions for each watertight compartment, and with connections to the steam fire and bilge pump in the engine room and to two 6-inch hand pumps, one forward and one aft.

DESCRIPTION OF MACHINERY.

General.—The machinery installation consists of one reciprocating engine of 1,800 indicated horsepower, two Scotch boilers burning fuel oil and the necessary auxiliaries. The equipment is modern in every respect and includes many points not found in commercial vessels of this class. The machinery arrangement plans and main engines are identical with those of the 150-foot Seagoing Naval Tugs *Allegheny* and *Sagamore*, built by the American Shipbuilding Company in 1918. The United States Steamboat Inspection rules and Lloyd's requirements are followed as is customary with this type of

naval vessel. It is to be noted that materials do not in all cases meet the General Specifications for Machinery of the U. S. Navy.

Main Engines.—There is one triple expansion, three-cylinder, inverted, direct acting engine, direct connected to a single right-handed screw propeller.

The cylinder diameters for all vessels of the class are, H.P. 21 inches, I.P. 33.5 inches and L.P. 55.5 inches with 42-inch stroke, designed R.P.M. 100 and steam pressure 185 pounds gage. All three piston rods are 5.25 inches in diameter. The order of cylinders from forward to aft is H.P., I.P. and L.P. The cylinder constants are as follows:

H.P. Cylinder	Top	.0367
	Bottom	.0344
	Mean	.0355
I.P. Cylinder	Top	.0934
	Bottom	.0911
	Mean	.0923
L.P. Cylinder	Top	.2565
	Bottom	.2542
	Mean	.2554

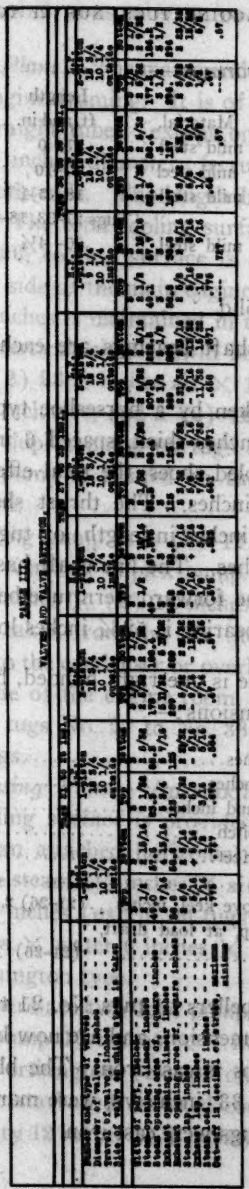
The data on valves and valve settings are as shown in Table III.

The valve gear is of the Stephenson type and is reversed by steam reversing gear fitted with differential motion. The cut-off is adjustable independently for each cylinder.

Connecting rod between centers 7 feet — 16 inches.

The ratio of the length of the connecting rod to the crank is 4.285.

Control.—The engine is controlled by a hand-operated lever throttle valve of the rotating disc type. This throttle, as installed, has worked with difficulty at times, the trouble evidently having been in giving the sides of the main and auxiliary valves too small a clearance, causing binding when heated, especially when in full open position.



Shafting and Bearings:—

	Material.	Length ft. and in.	Diameter inches.
Crank Shaft	mild steel	17– 0	11½
Crank Pin	mild steel	1–10	11½
Thrust shaft	mild steel	18– 5¾	11½
		(Tugs 27–33, 18–6)	
Stern Tube and Propeller Shaft	mild steel	20– 4¼	11⅝ (sleeve 13⅛)

All shafts are solid.

The six crank shaft bearings are each 13 5/16 inches in length.

The thrust is taken by a horseshoe type bearing having 7 collars, each 2¾ inches thick, spaced 6 inches. There are 6 babbitted water-cooled shoes, the total effective thrust surface being 840 square inches. The thrust shaft is supported by steady bearings 18 inches in length; on tugs No. 34 to No. 39, the length is 19 inches. The line shaft has one spring bearing 18 inches long. The forward stern tube bearing is 19⅞ inches long and the after bearing is 52¾ inches long.

Propeller.—There is one right handed, built-up, propeller of the following dimensions:—

Diameter, feet and inches	13– 0
Pitch as set, feet and inches	14– 9
Pitch adjustable, feet and inches	13– 9 to 15–9
Ratio of diameter to pitch	1.1346
Projected area, square feet	46.5
Disc area, square feet	132.7323
Height of lower tip above keel, inches	(21–26) 2, (27–33) 3⅝, (34–39) 11
Immersion of upper tip, at load draft, inches	(21–26) 17, (27–33) 34, (34–39) 8

The original propellers of tugs No. 21 to No. 26, inclusive, were of incorrect dimensions and are now being replaced. The material of the hubs is cast iron. The blades as supplied to tugs No. 27 to No. 33, inclusive, were manganese bronze, and those of the other tugs were cast iron.

Main Condensing Plant.—The main condenser is rectangular and built into the engine framing. It is of the surface type, 2 pass. with 1,221 straight tubes (except on 21-26 number of tubes is 1,227), ⅝ inches diameter, ferruled and packed in the tube sheets at both ends. The length between tube sheets is 15 feet 0 inches. The total cooling surface is 3,000 square feet (except on 21-26, cooling surface is 3,130). The condenser is on the port side of the main engine.

A vacuum of 27 inches is maintained in the main condenser by a Worthington, vertical, simple, twin beam air pump, size 14 inches steam × (2) 23½ inches air × 15 inches stroke on tugs Nos. 27 to 39, inclusive, and a (2) 12 inches × (2) 22 inches × 15 inches duplex pump on tugs No. 21 to No. 26, inclusive. The air pump is located on the port side of the engine room, forward.

The main circulating pump is a (2) 12 inches × (2) 18 inches × 18 inches, vertical, duplex pump (except on 27-33, the pump is (1) 12 inches × (2) 18 inches × (2) 18 inches. The suction may be taken from the sea, the bilge or the deck and the discharge is to the condenser or overboard. This pump is aft on the port side of the engine room. The above pump is N.Y.P.S. type on tugs No. 27 to No. 33, and Dean Brothers' type on other tugs.

Auxiliary Condensing Plant.—An auxiliary condenser of 250 square feet cooling surface is provided for port service. The condenser has an attached horizontal combined air and circulating pump, size steam 5½ inches × air 8 inches × water 8 inches × stroke 7 inches (except on tugs 21-26, size is 5½ inches × air 6 inches × water 6 inches × stroke 10 inches). All are of the Worthington type.

Feed System.—There are two feed pumps for the two boilers, one main feed pump 10 inches × 7 inches × 10 inches vertical, simplex, Worthington (except on tugs 21-26 it is Dean) located in the starboard forward corner of the engine room and one auxiliary 12 inches × 7 inches × 12 inches hori-

zontal duplex pump in the after part of the engine room on the starboard side.

The feed tank is located on the starboard side of the engine room at the forward end. A filter box is built into the upper part of the tank. The capacity of the tank is about 400 gallons.

A pressure, type closed feed heater is installed on the starboard side of the engine room near the feed tank; the type is as follows:

Tugs Nos. 21 to 26—No. 7 Reilly multicoil Type D, capacity 25,500 pounds of water per hour.

Tugs Nos. 27 to 33—Row and Davis, Horizontal, No. 22 paracoil 3-inch water and 5-inch exhaust connections capable of heating 35,000 pounds of water per hour to 200 degrees F.

Tugs Nos. 34 to 39—Row and Davis, vertical paracoil, size SO-1604.

Both the main and auxiliary feed pumps discharge through the heater to the boilers. The main feed pump has suction connections from the reserve feed tank, the main feed tank and the sea. The auxiliary pump has suctions from the same sources as the main pump and in addition has a suction from the boilers, the bilge and the ballast tanks.

A ¾-inch line is provided for taking make-up feed by vacuum suction to the main condenser from the reserve feed tanks.

Two 2-inch injectors are installed for feeding the boilers in port or in case of breakdown of both feed pumps at sea.

Miscellaneous Pumps.—A pump is provided for furnishing cooling water to the main engine guides and bearings and is located in the engine room on the port side forward. The size of the pump is listed in the pump table.

The pump for the sanitary system is located on the forward engine room bulkhead, on the port side. It has a suction from the sea only and a discharge to the sanitary system.

A fire and bilge pump is located in the engine room on the

port side on all except tugs No. 21 to No. 26, where it is located in same position as the next pump described. This pump has a suction from sea and bilge manifolds and a discharge to the fire main and the sea.

On tugs No. 27 to No. 33, inclusive, a general service pump is located in the extreme after end of the engine room on the starboard side, with a suction from the sea and the bilge and a discharge to the fire main and overboard.

On all except tugs No. 21 to No. 26 a small pump is located in the crew's quarters forward for filling the gravity tank from the ship's potable water tanks.

The sizes of the above miscellaneous pumps are as follows:

	Tugs 21 to 26	Tugs 27 to 33	Tugs 34 to 39
Engine Cooler Pump			
Type	Horizontal Dup.	Horizontal Dup.	Horizontal Dup.
Size	5½×3¾×5	5¼×4¾×5	5¼×3½×5
Sanitary Pump			
Type	Horizontal Dup.	Horizontal Dup.	Horizontal Dup.
Size	5½×3¾×5	5¼×4¾×5	5¼×3½×5
Fire and Bilge Pump			
Type	Horizontal Dup.	Horizontal Dup.	Horizontal Dup.
Size	7×12×12	14×10¼×12	12×7×12
General Service Pump			
Type	None	Horizontal Dup.	None
Size	.....	12×7×12	.....
Fresh Water (ford) Pump			
Type	None	Horizontal Sim.	Motor Driven
Size	.....	4½×4⅛×6	4×5 Recip.

The above pumps are Worthington on 27 to 39 and Dean Brothers on 21 to 26.

Steam and Exhaust Piping.—All steam piping except the main steam piping, is of copper. The main steam piping is seamless drawn steel. The exhaust pipes are of galvanized iron.

The main steam line is 6 inches in diameter. The leads direct from the two boilers join into a "Y" in the fire room and lead direct to the engine.

The auxiliary steam takes off from each starboard and port boiler with a direct stop, size 3½ inches, from which each line leads directly aft along the starboard and port sides of the engine room, with a 3½-inch cross-connection just aft of the engine room forward bulkhead. Branches from these two lines supply steam for the auxiliaries.

The auxiliary exhaust runs along the port side of the fire room and engine room with a circuit around the after end of the engine room leading forward to the auxiliary condenser and the feed heater, at the starboard forward corner of the engine room. This line is 5½ inches in diameter.

Boilers.—There are two Scotch, single ended, oil-burning boilers, one to starboard and one to port in the fire room, with the axis of shell fore and aft. The data for each boiler is as follows:

Designed working pressure, pounds gage	185
Test pressure, pounds gage	277
Draft	Howden, closed ash pit
Inside diameter, feet and inches	13– 3
Length, feet and inches	11– 1 15/16
Ordinary tubes	No. 321, O. D. 2 1/2 inches
Stay tubes	No. 75, O. D. 2 1/2 inches
Length of tubes, feet and inches	7– 7
Heating surface, square feet	2,485.8
Furnaces	Type, Morrison, No. 3, Diam. 40 inches
Furnace volume, each furnace, cu. ft.	108
Steam stop valves, inches	main 6, auxiliary, 3 1/2
Safety valves, inches	Duplex 3 1/2

Tugs No. 21 to No. 26, No. 34 to No. 39 have retarders installed in all tubes which, although promoting efficiency under forced draft, makes it impossible to operate under natural draft, without withdrawing the retarders.

Forced Draft Blower.—There is one forced draft blower located in the forward part of the engine room, supplying 10,000 cubic feet of free air per minute, through air

heater boxes installed in the uptakes of each boiler, to the furnaces using the Howden closed ash pit system. 2½ inches air pressure is needed at the blower discharge when the main engines are operated at full power. The blower fan is a 5½-A Niagara Conoidal, driven at 600 R.P.M. (rated speed) by a 5-inch × 5-inch engine. This blower has been generally unsatisfactory, as it is of too small capacity and of rather poor design for continuous duty. Many breakdowns have occurred, due primarily to the blower being run much above rated speed, and to the design of valve gear which causes frequent breakage of the valve stem. This blower has been replaced by a turbine-driven Terry blower on the *Wandank* and the *Montcalm*.

Uptakes and Smoke Pipe.—The uptakes from each boiler unite in the fire room and discharge to one smoke pipe, which has an area of 23.76 square feet and a height above the center line of the low (center) furnace of 49 feet 3¾ inches.

Fuel Oil System.—There is one fuel oil burner for each furnace or three burners per boiler. Tugs No. 21 to No. 26, inclusive, and No. 34 to No. 39, inclusive, are equipped with the White Fuel Oil System and tugs No. 27 to No. 33, inclusive, have the Coen System. Both systems use mechanical atomization and are supposed to operate at about 100 pounds, but 200 pounds pressure is actually used.

Each tug has two fuel oil heaters, those on tugs No. 21 to No. 26 and No. 34 to No. 39, having No. 2 Reilly, 40 square feet heating surface each, capacity 2,000 pounds oil per hour, and those on tugs No. 27 to No. 33, having capacity to heat 6 G.P.M. of oil from 60 degrees F. to 200 degrees F.

Two 6-inch × 4-inch × 6-inch horizontal duplex pumps supply oil direct from the tanks to the burners. Suction and discharge duplex strainers are provided.

Refrigerating Plant.—A combined refrigerator box and machine, of the type adopted for destroyers, is located on the superstructure deck. Data on these units is as per table V.

TABLE V.

	Tugs 21 to 26	Tugs 27 to 33	Tugs 34 to 39
Refrigerating Plant:—			
Rated capacity refrigerating effect	½ ton	½ ton	½ ton
Refrigerant	SO_2	Ethyl Chloride	CO_2
Manufacturer	Johns Manville	Clothel Co.	York
Refrigerated storage space, cu. ft.	58	73.6	67
Capacity drinking water, gallons	18	28	23.8
Type drive	motor	motor	motor
Distilling Plant:—			
Capacity evaporators g. p. day	2,500 approx.	2,700	2,800
Evaporators, number	1	1	1
Type	No. 6 Griscom-Russell	Bu. Eng. Vert. "U" tube	Row & Davis No. 22-6-C
Heating surface, sq. ft.	15.96	55.15	——
Distilling condensers, number	1	1	1
Type	No. 5 Griscom-Russell	Bu. Eng. Standard	Reilly No. 10-2CC type K
Capacity, gallons, day	2,500	1,450	
Evaporator feed pump, type	Horiz. Duplex	Horiz. Duplex	Horiz. Duplex
Size, inches	3×2×3	3×2×3	3×2×4
Evap. fresh water pump, type	Horiz. Duplex	Horiz. Duplex	Horiz. Duplex
Size, inches	3×2×3	3×2×3	3×2×3
Fresh water test tank	1	none	none

Distilling Plant.—The evaporator is operated on live steam only. Vapor may be discharged to the auxiliary exhaust line for make-up feed or to the distilling condenser for drinking and washing purposes. The distilling condenser has a capacity only about one-third that of the evaporator. The evaporator feed pump furnishes cooling water for the distilling condenser on tugs No. 21 to No. 26 and No. 34 to No. 39. On tugs No. 27 to No. 33 distilling condenser circulating water is furnished by either the engine bearing cooler pump or the sanitary pump. Data on the plant is as per table V.

Compressed Air System.—Tugs No. 21 to No. 26, inclusive, have one Westinghouse 11 inches × 11 inches × 12 inches air compressor with an air piping system to the fire rooms.

DESCRIPTION OF ELECTRICAL PLANT.

The generators and switchboard are located on a platform at the after end of the engine room on the starboard side. Two 10-K.W. D.C. turbo-generators are provided. They were manufactured by the Westinghouse Electric and Manufacturing Company, the data being as follows:

Voltage	125
Generator connection to turbine	direct
R.P.M.	4,000
Steam pressure	185
Steam consumption, full load, 0 lbs. gage back pressure	72.5 lbs./KW hr.

The generator exhausts to the auxiliary exhaust line only.

One switchboard is installed for the generator leads and one board for distribution to the various circuits.

The two-wire system is followed in the wiring for both power and light.

The motors installed are as follows:

Crew's ventilating	1 H.P.
Ice machine	2 H.P.
Fresh water pump (34 to 39 only)	1 H.P.
Radio	2 K.W.
Meat grinder	¾ H.P.

Two low-power fixed 18-inch searchlights are installed. One 12-inch portable low-power searchlight is also provided.

The radio installation consists of one 2-K.W. spark set and one Bureau Engineering No. 143 short-wave receiver. All vessels are also being fitted with vacuum tube detectors, type C.F. 122.

PROGRESS IN PROPELLER DESIGN.

By S. E. Slocum, Ph. D.,* Associate Member.

As the marine propeller came into existence only 75 years ago, and the aircraft propeller is scarcely 25 years old, it is to be expected that there should be numerous points in propeller design which have not yet been fully settled.

Evidently propeller action must follow certain definite universal laws whether the propeller works in water or in air. The limitations of present propeller theory are therefore clearly apparent in the fact that the method used in designing marine propellers and the method used in designing air propellers bear practically no relation to each other. In fact, a designer who has had experience in both fields recently remarked to the writer that a man who was experienced in designing marine propellers would be very poorly equipped to design an air propeller. This indicates that there must be some vital discrepancy in the underlying theory, as the laws of mechanics are universal in their application.

In spite of these defects in theory, marine propeller design has been almost completely standardized in practice, and follows certain prescribed routine methods. The starting point in marine practice consists in the information furnished by the performance curves of full sized propellers of certain standard types working behind a ship. Thousands of such tests have been made and by the method of comparison and elimination, certain types of propellers have been arrived at, of which it can be said that they are more efficient under the conditions of the test than any others tried. Starting with this experimental data, marine propeller design proceeds by changing one element at a time, such as diameter, pitch, or projected area ratio, and comparing the performance of the new design with that of the original from which it was modi-

*Consulting Engineer, General Machinery Foundations Co., Philadelphia, Penna.

fied. This method shows that certain propellers are better (or worse) than others, and thereby enables an intelligent selection to be made from the types at our disposal, namely from the types which so far have been experimentally tested.

In this method, however, certain important factors are left entirely out of account, or determined arbitrarily, chief of which are the profile of the blade and the form of its cross section. The blade form commonly adopted in marine practice is that of a "true ellipse", although no valid reason for adopting this shape has ever been advanced except that it is symmetrical and pleasing in appearance. As no one would ever think of designing an air propeller on these lines, it may be inferred that there is no real reason why it should be taken for granted that an elliptical shape of blade is specially adapted for propulsion.

The same idea of symmetry is followed in determining the blade cross-section, which usually has the form of a lune or circular segment. This design of course has the merit of simplicity, but otherwise has no rational basis. In fact there is reason to believe that the point at which cavitation occurs is largely dependent upon the form given to the trailing edge of the blade. To say the least, therefore, there is no evidence that the present forms of blade profile and cross section as adopted in marine practice are the most efficient that can be devised, while on the contrary a rational method of design will indicate what improvements are possible and practicable.

As indicated above, marine practice in propeller design is based almost entirely on experiment, and very little theory is involved. Consequently if there is a decided change in the type of hull, thereby altering materially the wake and thrust deduction factors, or in the speed it is desired to attain, such for example as in developing a type of vessel of double the speed of existing types, the designer has very little data on which to base his calculations, and must depend entirely on the model tank and the law of similarity, which is not always entirely satisfactory.

The construction of air propellers, being entirely outside of the field of marine experiment and practice, brought about the elaboration of a theory of propeller action in the air. At present the design of air propellers, both in this country and in England, is based on the so-called "impact theory" originated by Drzewiecki, and modified by the "inflow theory" proposed by Henderson. The results of analysis by this theory, however, vary considerably from results obtained by experiment, and are therefore not relied upon to give the absolute elements of design, but rather to show the characteristic features of certain types of propellers. Although a step in advance of marine practice, this method is also susceptible of great improvement.

In a recent book on air propeller design by a leading English authority, a numerical example is given to show the wide variation in results which may be expected from the impact theory. In this problem it was required to design a propeller for an aeroplane fitted with an engine developing 250 H.P. at 1200 revolutions per minute for a flying speed of 110 miles per hour. Assuming a propeller diameter of 10 feet and proceeding with the design in accordance with the impact theory, neglecting inflow velocity, it was found that the propeller so determined would absorb 280 H.P. at 1200 revolutions per minute with an overall propeller efficiency of 84.7 per cent. Redesigning this propeller, using exactly the same numerical data but including inflow velocity, it was found that this new design would absorb 182 H.P. at 1200 revolutions per minute with an overall propeller efficiency of 65 per cent, a difference therefore of about 100 H.P. in the power absorbed and of 25 per cent in the overall efficiency. In discussing this discrepancy the designer states that if this propeller was tested in actual flight it would probably absorb about 260 H.P. at 1200 revolutions per minute with a thrust efficiency of 80 per cent.

It requires no mathematical analysis to understand some of the reasons why the impact theory yields results like the

above. There are at least two well established facts which may serve to indicate where the trouble lies. The first fact, which has been established by innumerable tests on aerofoils in wind tunnel experiments, is that the "lift" of an aerofoil is due mainly to decrease of pressure on the upper, or suction, side, not over one third at most, and usually much less, being due to pressure on the lower face. Evidently, therefore, the impact theory as ordinarily understood and applied is insufficient to account for the lift of an aerofoil, or the thrust of a propeller. Furthermore, it is also a well known fact that when a propeller is revolving at the speed of revolution and with the speed of advance for which it was designed, the angle of incidence of the relative flow with the pressure side of the blade is frequently zero, or at most a very small angle, say of about 1 degree. From this fact also it is apparent that the impact theory is inadequate to account for propulsion.

In order to develop a theory of propeller action which shall be both practical and scientific, it is necessary to start from a hydrodynamical basis, as any theory which is not in accord with the laws of fluid flow is evidently unsound. Numerous investigations have been carried out in recent years along purely hydrodynamical lines and these results are now available for establishing propeller design on a rational basis. In fact, the circulation flow around an aerofoil, established by the investigations of Kutta, and which is now generally taken as a basis for calculating the sustaining power of airplane wings, has provided a means whereby a propeller theory in accordance with hydrodynamical laws can be developed. This circulation theory has been elaborated to a considerable extent abroad, and is already established on a practical working basis. Undoubtedly the next few years will see the circulation theory made generally available in this country, as marine engine practice made great strides during the war, and anything that will better propeller efficiency will not long be overlooked.

As the idea of circulation flow around the blade is the distinctive feature of the modern propeller theory, an effort will be made to explain briefly just what is meant in hydrodynamics by circulation.

In order for a propeller to function, there must be a difference in pressure on opposite sides of the blade. On the leading face there is an increase in pressure, whereas on the rear face there is a decrease in pressure. Suppose now that a tube open at the ends is bent into the form of a C and placed so as to partly encircle the propeller blade, one end being adjacent to the leading face and the other end adjacent to a point directly opposite on the rear face of the blade. Evidently the difference in pressure at the two ends of the tube will then cause a circulation, or flow, of the fluid in the tube. If the velocity of flow set up in the tube is v and the length of the tube is l, the circulation through the tube will be vl.

Now instead of an actual tube consider a closed curve intersecting the blade, or completely encircling the blade and intersecting the race stream just in the rear of the blade. Then the line integral of the velocity around such a closed circuit is called the circulation, and represents an actual circulation flow around the blade as in the case of the bent tube used for illustration. Denote the density of the fluid by δ and the pressures on opposite sides of the blade where it is intersected by the closed circuit under consideration by p_1 and p_2. Then the difference in pressure, $p_1 - p_2$, will produce, in the time t, a circulation C of amount

$$C = \frac{p_1 - p_2}{\delta} \cdot t.$$

If the velocity of advance of the propeller is denoted by V and the breadth of the blade in the direction of advance at the point considered is b, then if t is the interval of time during which the circuit continues to intersect the blade we

have $b = Vt$, or $t = \frac{b}{V}$. Consequently the expression for circulation may be written as

$$C = \frac{p_1 - p_2}{\delta} \cdot \frac{b}{V}$$

But $(p_1 - p_2)\, b$ is evidently the thrust (or lift in case of an aerofoil) exerted on a unit length of blade surface. If we denote this thrust per unit length by T, we have

$$T = C \delta V$$

This is the famous circulation theorem discovered independently and in different ways by Kutta in Munich and Joukowsky in Moscow.

In order to make clearer the effect of circulation currents on the flow around an arched surface, the accompanying diagrams are reproduced from an article by Dr. Daimler. The cross section of the surface as shown in the figures is an arc of a circle with a camber of $\frac{1}{12}$. The length of the surface is assumed to be infinite in order to avoid the disturbing effects at the ends, and also to simplify the calculations by reducing the problem to the case of two-dimensional flow. The surface is assumed to be moving through still air with a velocity of 20 meters per second, at an angle of attack of 10 degrees. The curves shown in the accompanying figures were actually calculated from the given numerical data, and not merely drawn by guess or in a qualitative manner.

For purposes of analysis, the flow was separated into three elements: chord currents (Fig. 1), normal currents (Fig. 2) and circulation currents (Fig. 3). The first two of these partial currents are combined in Fig. 4, and give the flow which avoids the obstructing body in the easiest way, and as is well known, produces no thrust and offers no resistance to motion (zero lift and drag in the case of an aerofoil). The combination of all three elements gives the flow shown in

Fig 5. The fact that the stream lines are crowded together on the convex side of the surface indicates an increase in velocity in this region and a consequent drop in pressure, while the reverse is true for the concave side.

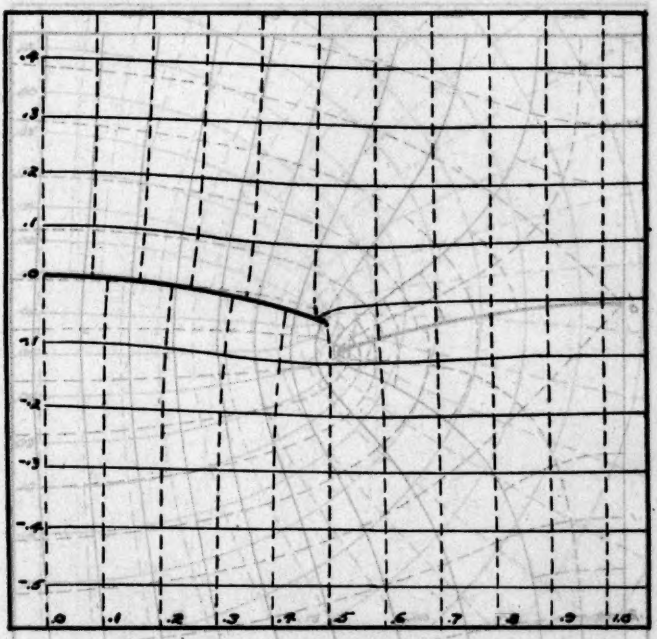


FIG. 1.—CHORD CURRENTS.

Besides giving the magnitude of the resultant force on an elementary area as expressed by the above simple formula, the Kutta-Joukowsky theorem also states that the direction of this resultant is at right angles to the direction of flow at infinity, which means practically at such a distance ahead of the object as to be free from the disturbance created by its motion through the fluid. This fact in regard to the direc-

tion of the resultant accounts for the high efficiency of a propeller (or aerofoil) when the angle of incidence is very small.

A rational theory of propeller action such as that based on the circulation theory affects propeller design in a number of

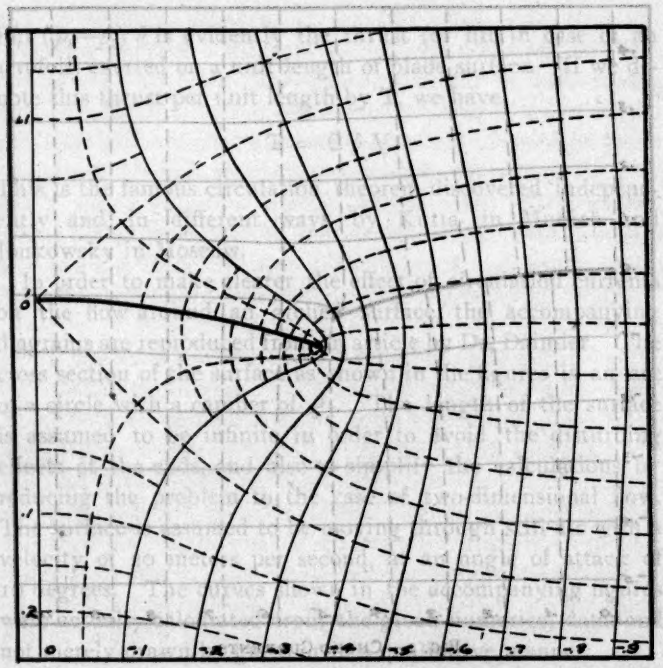


FIG. 2.—NORMAL CURRENTS.

ways. In the first place it enables the designer to lay out a propeller to meet any *given set of conditions*, with assurance that the predicted performance of the propeller will not differ materially from actual results in service. Also the same methods of design apply to both marine and aircraft propellers, as the laws of hydrodynamics are independent of the

medium except for the numerical values of the density and viscosity coefficients involved. A notable feature of the circulation theory is that it makes it possible to develop a type of propeller for maximum efficiency. The importance of

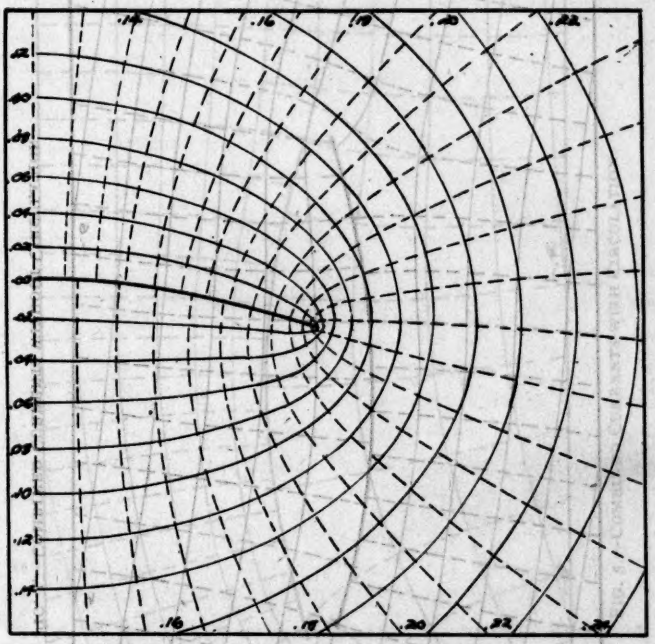


FIG. 3.—CIRCULATION CURRENTS.

such a type is well recognized in aeronautics, while in marine practice, where a vessel moves at a uniform speed for long periods of time, even a slight betterment in efficiency is a notable achievement from the standpoint of power saving alone. Elements of design such as blade contour and cross section, which are now purely empirical in marine practice and largely so in aeronautics, may also be determined with

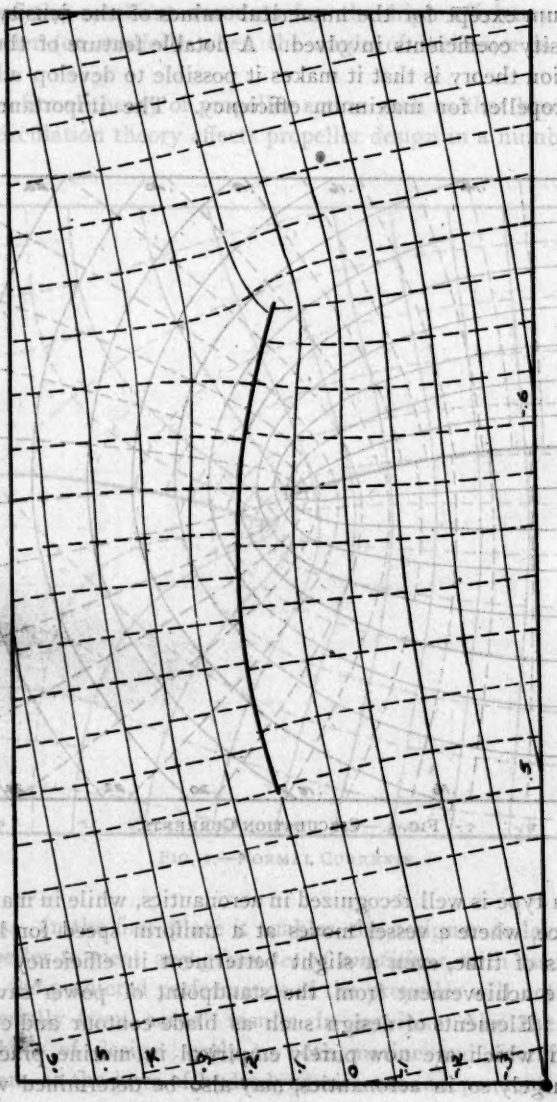


FIG. 4.—COMBINED CURRENTS WITHOUT CIRCULATION.

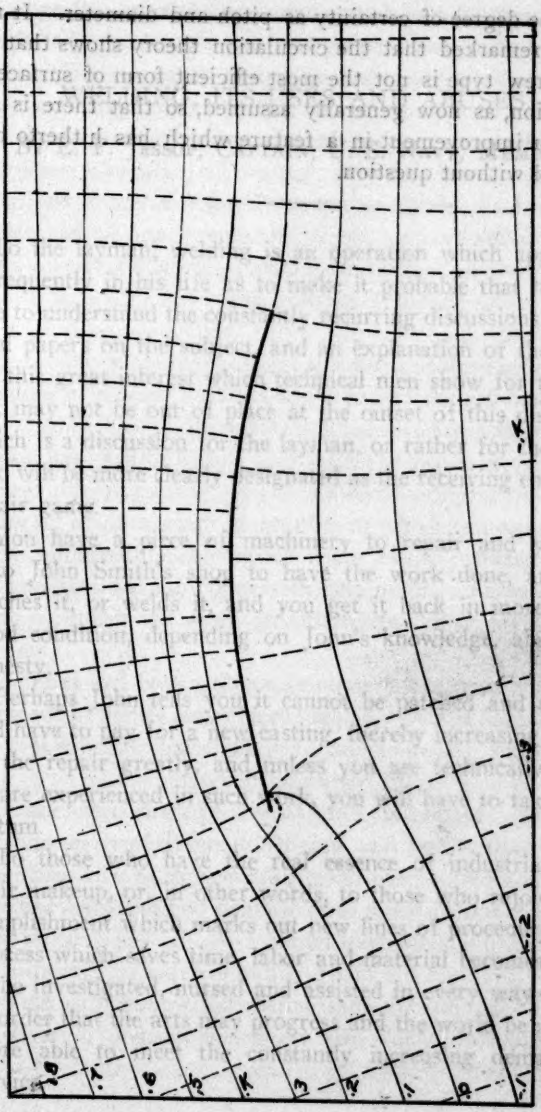
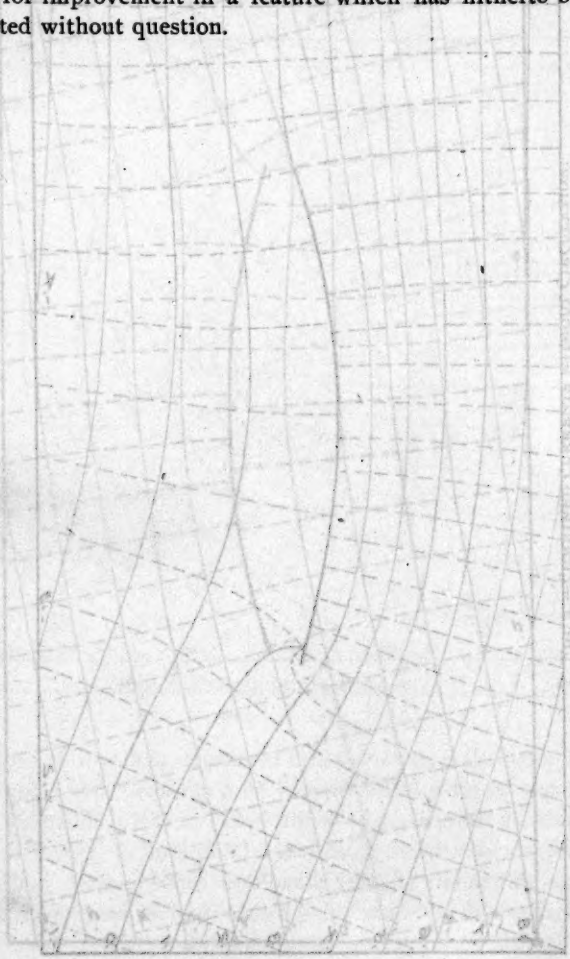


FIG. 5.—COMBINED CURRENTS WITH CIRCULATION.

the same degree of certainty as pitch and diameter. It may also be remarked that the circulation theory shows that the true screw type is not the most efficient form of surface for propulsion, as now generally assumed, so that there is also room for improvement in a feature which has hitherto been accepted without question.



WELDING; ITS USES AND ABUSES.

By E. P. Jessop, Captain, U. S. Navy, Member.

To the layman, welding is an operation which appears so infrequently in his life as to make it probable that he is unable to understand the constantly recurring discussions in technical papers on the subject, and an explanation of the reason for this great interest which technical men show for this subject may not be out of place at the outset of this discussion, which is a discussion for the layman, or rather for the person who will be more clearly designated as the receiving end of the repair game.

You have a piece of machinery to repair and you take it to John Smith's shop to have the work done, and John patches it, or welds it, and you get it back in more or less good condition, depending on John's knowledge, ability and honesty.

Perhaps John tells you it cannot be patched and that you will have to pay for a new casting, thereby increasing the cost of the repair greatly, and unless you are technically trained or are experienced in such work, you will have to take John's dictum.

To those who have the real essence of industrial life in their makeup, or, in other words, to those who rejoice in accomplishment which marks out new lines of procedure, a new process which saves time, labor and material becomes a thing to be investigated, nursed and assisted in every way possible, in order that the arts may progress and the world be made the more able to meet the constantly increasing demands for service.

Welding with the electric arc is today in its infancy when compared to its possibilities, and it is the feeling that this is true which makes technical men everywhere talk and write of welding.

This is not saying that the layman is not interested in saving time, material and labor, because he is the most vitally interested of all, but he cannot usually devise means and methods, as that is the province of the expert, and he has neither the time nor the inclination to investigate sufficiently to know which methods should be adopted since he has the expert to do that for him.

The purpose of this paper is to discuss welding without technicality and in such a manner as to interest those who probably need it in their business but don't know it.

Welding as an art is a family of many brothers and sisters, and like all families where visions of parental wealth appear, the discussion as to the proportion of this wealth to be willed to each member causes much acrimonious interchange, with the usual result of disrupting the family, discrediting the individuals in the family and frittering away the wealth in prospect.

This has been exactly true of welding, and has resulted in discrediting all types of welding to a certain extent in the mind of the layman, and in making even the technical man wary of a too solid backing of any particular method.

Thermit fought Oxy; Oxy fought Electric, both Spot and Arc, while Spot and Arc fought among themselves, and the prospective client, after listening for a while to what each said about the other, in many cases decided they were all bad and went back to old methods of repair.

Even worse than the above, different types of Arc discredited other types of Arc, until it is small wonder that so little real information which may be depended upon is available for the layman to study.

The art has suffered from another evil, which time and competition only can eradicate, and that is incompetence in the personnel of the various companies doing work. This element is most pronounced in the Arc world, since the Thermit and Oxy members are older and, therefore, more completely settled on a practical basis.

Still another confusing element for the layman has been the fact that the greater part of the discussions written on the subject, especially in the Arc Weld division, are written entirely from a technical standpoint, until one almost expects the microfarad to materialize and to be able to distinguish him actively supervising the depositing of the globule of metal in the weld.

In reality welding is entirely a practical problem and its "goodness" or "badness" is directly dependent on the mechanics of the application by the personnel much more than it is on any other agency, but this idea is not usually prominent in the technical discussion.

The situation, as it appears to the writer, is about as follows as regards the types of welding and their fields of usefulness:

Use Thermit where the work can be gotten at from all sides and where there is a mass of metal to be added.

Use Oxy-Acetylene for cutting and for welds which do not have to stand fluctuating or vibrating strains in all metals except cast iron; where pre-heating can be properly carried out without seriously affecting the time and cost of the job, and where intense heat cannot injure other parts of the mechanism; also where hard metal will not be a serious drawback; never on a working surface nor in a place where ductility is needed in the weld.

Use Arc welding (high or low voltage) where medium tensile strength and ductility are a criterion and the weld is in steel or any other elastic metal of good tensile strength, also in

alloys if the machine is so arranged as to permit the expeditious reduction of voltage to give a proper welding heat.

Use Arc weld (low voltage) for work in cast iron, alloys, or steel where high tensile strength and ductility are necessary, and especially where vibration is to be met in service; also where the work is to be repaired in place so that pre-heating is difficult, since the strains set up in the work by cooling will be negligible where the low voltage arc is used, and with it pre-heating is not necessary; also where the weld is part of a working surface and where soft metal is a requisite.

Good welding is possible only where the proper welding heat is maintained constantly in the weld. In the blacksmith and forge shop the successful weld is a matter of the trained eye of the blacksmith for heat color, while in electric arc welding it is in the setting of the heat control adjustment to the work and the ability of the control to maintain the setting within very close limits.

It is by no means sufficient that an energetic salesman tell you that his type of apparatus will perform all kinds of work, nor can you depend upon the literature passed out to you by the corporations which manufacture welding equipment as to the ability of a particular type of machine to do certain kinds of work. Such corporations have not been and are not now careful in their statements as to what their output will do, and it is in that circumstance that lies, the writer believes, the major part of the difficulties by which Arc welding has been kept from coming into its own.

Corporations which would scorn to mislead with regard to equipment in other lines and which appreciate the necessity in an advertising way of having equipment fulfill the claims made for it, do not hesitate, in the case of welding equipment, to "draw the long bow," and the writer explains this rather curious fact by the probability that so little is known with regard to practical welding by the sales force, the art being

so young as to make even the practical side somewhat nebulous as far as established practice is concerned, they assume all types of machines, metals, etc., and all other adjuncts of making a weld as being without essential differences, whereas the truth is there is the greatest difference in the world.

Certain standard builders of such equipment have constantly maintained that cast iron cannot be reliably welded with the electric arc, and yet they will readily sell their equipment to those who need to weld cast iron, just because another firm will guarantee to successfully weld cast iron, and they feel in a vague way that while they do not believe that their machine will do that work, yet if the other firms' equipment will, why not theirs.

The sooner this attitude is abandoned, the sooner real welding will become a standard method of repair and be put in the relative position with regard to other methods that its importance requires. In other words, the commercial side of welding has been permitted to run wild, with the inevitable result of instilling in the minds of prospective clients a suspicion which is a direct block to proper growth in the art.

During the welding which put into service that great fleet of interned German ships in the incredibly short time of five and one-half months against an estimated time of eighteen months to two years, a desperate attempt was made by one arc welding equipment company to prevent the use of arc welding on the cast iron cylinders of the engines of these ships, by reporting the writer to higher authority for the use of dangerous methods of repair "since electric welding experts such as themselves knew that cast iron could not be safely welded," and yet one year later when the world knew that cast iron had been successfully welded and that 162 great welds in these cylinders had stood the test of war operation without a single weakness developing, the same firm of electric welding experts in full page advertisements celebrated the welding of the en-

gines of the German ships with the electric arc and announced that they sold such equipment, yet at the time of issuing this advertisement their machine was not one whit better designed to handle cast iron than it was when they, honestly, I think, stated that such welding could not be done.

If their advice had been followed, four hundred thousand men who faced the Germans in the final months of the war from Chateau-Thierry to Sedan would not have been in Europe, and electric arc welding would have been immeasurably set back by the loss of this, the greatest piece of advertising the art has ever had.

It must be definitely set down here that cast iron has been and can always be reliably welded by the use of open circuit low voltage equipment with automatic heat control, and that it cannot be reliably welded with open circuit high voltage equipment without automatic control, and it would assist the general health of the art if this fact were realized by those who have for so long obscured the situation for commercial reasons, and an appreciation shown of the fact that certain types of equipment are definitely limited in their scope while others are not so limited.

The electric welding world is its own worst enemy in very many ways and it does seem a deplorable thing that such an aid to time and money saving should be forever having its progress blocked by overzealous sponsors.

From time immemorial it has been the mark of a good salesman that he appreciate the good points of his competitors' product and if he could not show that his own was honestly as good or better he kept quiet on those points or, if requested, would give his honest opinion, but this has not been true of the Welding World where cut-throat tactics have been the rule rather than the exception, and, as stated before, the prospective purchaser has been frightened away by what each had to say of the other's apparatus, and has been made to believe that

all welding is unreliable and a makeshift when in reality there is no form of repair more reliable or lasting when properly done.

Anyone who reads the various pamphlets issued by the various firms, and who knows welding with the electric arc or with oxy-acetylene, must find great amusement in the loose and misleading statements therein and the naive way in which a statement will be made on one page when talking on the selling side and contradicted on another when talking on the operative side, and all within a page or two of each other.

One is also struck with the great amount of vague general information with regard to practical welding and the almost entire lack of specific information on that subject. As a case in point, I quote from a pamphlet issued by one welding equipment firm under the heading of "contraction of the hot deposit": Question, "How may strains produced by this characteristic be minimized?" and the wise answer, "By adopting a proper welding procedure, either non-rigid or rigid, which serves to partly anneal the metal and reduce the locked-in stresses." To adopt the parlance of the old sailor man, "By all the Gods of War," what does that mean to a practical welder? Not a word of how to apply the metal so as to prevent the formation of these stresses, but a lot of vague claptrap which any practical man would throw into the waste basket in disgust.

The truth is that the electric experts have been spending their time in inventing nomenclature to such an extent that they have forgotten to watch a good operator sufficiently long really to find out how he does obviate stresses, hard metal, burnt deposit, etc., and since it is the electrical expert who writes our welding dope we are fed up with generalities when we need particular information.

The writer had one experience with an "expert" in welding which was illuminating. The expert in question was a pro-

fessor in one of our large universities, whose business it was to lecture on welding. He stated to the writer that an electric arc weld in cast iron could not have the strength in it to make it reliable, and when questioned as to why, he said that the stresses set up by contraction in a metal so lacking in ductility as cast iron were so great as to preclude placing much dependence on the weld. When shown by sketch how the stresses of contraction could be so minimized as to make them negligible, he agreed that it could be done and said he had never thought of that.

The old saying that a little knowledge is a dangerous thing was never more strongly exemplified than in the case of welding, and any man who presumes to talk welding without a knowledge of the practical as well as the theoretical side is treading on dangerous ground unless he definitely states the exact limitations of his observations.

Welding is dependent on the following for its characteristics:

Type of apparatus used.
Kind of welding metal used.
Ability of the supervisor.
Ability of the operator.

Defective welds may be defective in the following ways: Internal stresses set up in the work by improper equipment and method; metal not thoroughly fused due to too low heat; metal burned or rendered brittle by too high welding heat; weld slagged up by careless operator not cleaning surface of each layer of metal and by fluctuating heat in the weld, or the too frequent breaking of the arc by a careless operator, especially when using a high voltage machine (anything over fifty volts).

To reduce the probability of these various defects creeping into a weld, the operator must be trained, of course, but there are various ways by which the type of machine may assist in

reducing this probability, and also by which the kind of metal used for welding and the kind of supervision provided may also assist.

The first item, i. e., stresses in the weld, is not only a matter of supervision and operator, but also the type of equipment used.

The laying out of the scheme for the work must take into consideration so applying the metal as to reduce the generation of stresses in the finished work to a minimum.

The amount of heat used in making the weld is both a function of the type of machine and the good sense of the operator; the type of machine, because certain types permit a wider range of heat after being adjusted for the work than other types, and a function of the operator, because he has it in his power to reduce the amount of heat to which any part of the structure is subjected, both by holding a steady arc and by shifting his point of work sufficiently to keep the whole mass comparatively cool. The speed of welding is also in his control.

The adjustment of the machine to the work so as to get a proper welding heat is distinctly a function of the operator and he must be sufficiently expert to judge when the metal really is being welded. Carelessness in this regard at the beginning of the work is frequent cause of poor welds.

The burning or the deposit of hard and brittle metal is a combination of the operator and the machine since the drawing of a long arc will throw the whole heat scheme out of adjustment temporarily and the combination of an inefficient or careless operator with a machine which permits the drawing of an arc longer than usual will insure a bad weld, and no machine which does not have close control of the current and which carries an open circuit voltage of greater than fifty volts can be considered as certain to turn out as reliable welds as one which has these two features. In other words, it takes a much more expert and conscientious operator to make contin-

uously good welds with a high voltage machine than it does with one of low voltage in spite of the fact that the low voltage arc will break more frequently than the high voltage, but the breaking of the arc in the case of the low voltage machine is not attended with slagging up or burning the metal of the weld as in the case of the breaking of a long arc, and the outside limits to which the low voltage arc may be drawn before breaking are not outside the limits of proper welding heat, while those of the high voltage arc are.

Annealing is not necessary in a properly made weld with a low voltage. It is necessary with high voltage and is an essential in the case of an oxy-acetylene weld except in cases of simple filling of blowholes where strength is not a consideration and where the holes are not of great extent.

Oxy-acetylene welding is accompanied by many difficulties which do not exist to a serious degree in arc welding. For instance, the great heat of the flame makes it impossible to keep the work cool and therefore pre-heating and subsequent annealing becomes essential. This precludes doing work in place as so frequently can be done in arc welding. Again, the great heat and other characteristics make it difficult to re-weld parts which are found defective without re-welding the whole work, whereas any part of an arc weld may be cut out and re-welded without disturbing the rest of the weld.

The fact is, a properly made arc weld is made by laying in the metal and then by the hammer test developing the hard spots and those portions in which the metal has not fused properly, and cutting out those spots and laying in new metal. As a case in point, we had to cut out about three inches of an eight-foot weld in one of the turbines of the *Vaterland* and renew it, those three inches being the only part of that great weld which was not perfect, all of which was determined by the hamer test on inspection of the weld after the operators had completed it, and before any test was put on the turbine.

With regard to the arc systems, the work will heat up more with the high voltage than with low voltage arc and as a result the stresses will be that much more in extent, which is the reason for the statement above that there is no reason for preheating and subsequent annealing with the low voltage but there is some for the high voltage.

It is seldom, for instance, that an Oxy or a high voltage weld will show more than three or four per cent elongation on test while the average low voltage will show between ten and twenty per cent with the same welding wire.

If the welding world is to be put in its proper place of importance in the industrial community, it must desist from its present method of welding anything with any equipment, because the percentage of failure is so great as to throw suspicion on the whole art, and in truth there is no reason why a guarantee should not be required on something like the "no cure, no pay" basis.

Any competent welder should be able to look at a job and say whether the outcome of a weld would be uncertain and in such a case he should not attempt to work without telling the client of the chances being taken and giving him the opportunity to use other methods if he so desired.

While it is not claimed that all companies in this field are careless as to their statements to clients and as to the results of their work, there certainly is a great deal of poor welding being foisted on the public, and since the art is not yet sufficiently standardized as to be received with entire confidence, such carelessness cannot but seriously affect the expansion of the art very detrimentally.

At the outset of the use of the art, only those jobs were considered electric welding jobs where strength was not too much of a requisite, and as a result, any old method of laying on metal would suffice. This caused carelessness among the operators and a lowering of requirements among the supervisory

forces of the various companies in this field, so when welding expanded into the field where the original strength of the metal was required in the weld, there were few operators and supervisors who were capable of meeting the new requirements successfully.

This was the basic reason for the condition early in 1917, when the railroads had to furnish the welding talent for the repairs to the German ships, since the railroads had been for two or three years using the arc in their own repairs, and naturally they had to have those repairs reliably made, so that their personnel was taught proper methods in welding and to a great extent were forced into a feeling of responsibility for their work.

Just the opposite was true of the marine world where repairs of such a character were made by commercial firms whose interest was in getting the work done in the time allowed and at a figure which represented a profit for them. What happened to the poor engineer when welds let go at sea never disturbed the peaceful slumbers of the ship repair man, the result being that the railroad fraternity went ahead in their knowledge of welding while the marine world slept.

The application of welding to the marine industries has yet to be worked out to its logical conclusion and in that there is much to be accomplished by educating the world to a proper belief in the efficacy of the method, and this education cannot be successful if the welding world continues content to discredit the art by careless or questionable application.

The field of welding lies in the home, in the shop, in the foundry, in the whole gamut of industry, and on the broad expanse of the sea. No shop and no ship is completely equipped without its welding equipment and its capable operator, and the reason is that it is the greatest saver of time, material and labor the industrial world has seen in many years.

At sea it has a tremendous value as there are many breakdowns of machinery which more or less totally disable ship-

ping, the character of which makes it easy to repair with a proper welding equipment, and frequently one such repair in two or three voyages would affect a saving in time which would pay for the equipment many times over.

Much is being written on the subject of the difficulties of testing welds sufficiently to assure their reliability.

The writer's experience has been very reassuring in that regard and it is believed that more extended experience with actual work by the writers of articles on welding and by those who have need of the art in their business will eradicate much of this idea.

The writer has proven to his own satisfaction that the hammer test, which by the way is seldom mentioned as a method of testing a weld, will invariably show up a really defective weld; that is, a weld which is sufficiently defective to seriously affect its strength.

Just as you can detect laminations in metal by the hammer, so you can detect unwelded spots in the weld, and by the action of the metal under the hammer you can determine its ductility and general condition.

It is seldom that a pressure test will develop a weakness in a weld which will not be shown by the seepage through of the liquid before the test pressure is applied. Weld a crack in a cylinder; fill it with water and if the water does not come through without pressure it seldom will appear when the pressure is applied because a weld which is defective to a point of weakness will not hold water. This, of course, is not an infallible rule because in a weld of any thickness it is possible that a wall impervious to water might have been formed while the greater part of the metal did not adhere, but this is not a probable happening, and would be shown up by the hammer test.

In other words, you may trust a weld which does not show defects by visual inspection and the hammer test, both of which are practical tests for any work.

The ordinary porosity of the welded metal should not be confused with bad welding because it seems not to weaken the structure to an appreciable extent and where work must be water-tight the peening of the surface is all that is necessary to prevent seepage.

The welding metal to be used is another item which should have careful attention, as it is obvious that any old metal will not make good welding metal.

Several firms have carried out extensive laboratory investigations on this subject and from them, welding metals which have the proper proportion of the various elements which go to make up the characteristics of the metal being worked on may be obtained. Passing through the arc burns out percentages of the various elements and it is therefore necessary that the welding wire contain a calculated excess of the elements which burn out in order that the metal after passing through the arc will still retain proper proportions of these elements.

Therefore, one who is going to engage in welding operations should investigate the metals just as carefully as he investigates the characteristics of the machine he buys.

Last, but by no means least, comes the supervision of the operators. Much has been written of the importance of the operators, but of immensely greater importance is the man who lays out the work and passes on it as it proceeds. It is a well known fact that personal pride will make any operator tend toward being a biased critic of his own work, and he therefore is not ordinarily of much use in making a decision on the efficiency of his work. Too little supervision is one of the crying evils of the welding game today, and one which must be corrected by the welding world itself before dependence will be placed in them by the users of their product.

It is appreciated that some of the above sounds a bit critical and fault finding, but the reader may rest assured that the intent is to help and not to hurt, and the seriousness of the

criticism comes from a sincere conviction that the world is not today receiving more than a small percentage of the assistance it should receive from the welding world, and the welding world is not establishing itself in its proper place in the industrial world due to the various items of bad practice mentioned herein.

TEST OF A COCHRANE OPEN FEED WATER HEATER.

By Lt. P. J. Searles (C. E. C.), Member.

In connection with the extension of the Power Plant at the Naval Proving Ground, Indian Head, Md., an extensive test was made of a new Cochrane open feed water heater, the results of which are embodied in this article. As the method used was different from the usual scheme, and does not require the apparatus needed for the usual testing, its exposition may be useful. The results, which are not as accurate as those given by a complete and thorough laboratory test, are close enough for all practical purposes. The location and piping of the four heaters installed are shown on the attached drawing, although but one heater was used in the test.

The principal object of the test was to determine the capacity of the heater when supplied with steam at two pounds per square inch gauge, and heating the water from 60 degrees to 210 degrees F. The rate of supply (taken from a stand pipe and several degrees above 60 degrees F.) could not be closely governed as the heater tested was in service, and boiler feed was taken from the heater at a varying rate. Also a steam pressure of 1¼ pounds was the maximum that could be obtained with satisfactory steadiness at the time, although the exhaust steam used did possess superheat in all except one test. It was therefore necessary to test under prevailing conditions, and devise a method of translating results into terms of the desired conditions which will be reached when the plant

is in complete operation. The most satisfactory method and formula seemed to be as follows:—

$$\text{B. T. U. supplied to water in one hour} = K\,(V)^{0.6}\,\frac{d}{d-t}$$

Where K = coefficient of heat transfer from steam to water per cubic foot of water.

V = Quantity of water supplied to heater in one hour in cubic feet.

d = Logarithmic mean temperature difference between steam and water in degrees F., that is,

$$d = \frac{t_2 - t_1}{\log_e \frac{t_s - t_1}{t_s - t_2}}$$

Where t_2 = Temperature of discharge water in degrees F.

t_1 = Temperature of intake water in degrees F.

t_s = Temperature of steam in degrees F.

Presumably when the actual conditions approximate those specified above, but slight error would result from the use of this formula. In the tests which were relied on principally to fix the value of the maximum obtainable capacity of the heater, conditions were very close to those specified. For example the mean data derived from test No. 18, was as follows:—

Back pressure = 1.20 pounds per square inch gauge.

Temperature of supply water = 60.2 degrees F.

Temperature of discharge water = 208.2 degrees F.

Mean temperature of steam = 216.2 degrees F.

(Note: The temperature that would correspond to a steam pressure of two pounds would be 218.5 degrees F.)

The superheat found in the steam supplied to the heater would not introduce any appreciable error into the results, for the degree was comparatively slight. If the steam had not been superheated an uncertainty would have existed as to the amount of priming introduced with the steam.

It can be seen that from the measured back pressure and the degree of superheat, there has been calculated the mean temperature of the steam in each test. It is realized that the back pressure, as measured, was not the partial pressure of the steam, but no methods were available for distinguishing between the total pressure of the steam and air present, and the partial pressure thereof. This fact may be considered as somewhat disadvantageous to the heater, but this handicap would decrease as the performance of the heater improved, for the ability of a heater to free itself from air is about the largest factor contributing to the performance of the heater.

A series of tests were made at different back pressures, and the "K" and quantity of water supplied to the heater per hour were calculated for each test. These values are plotted on Chart No. 1. Curves were then constructed on this chart for back pressures of ¼, ½, ¾ and 1¼ pounds pressure and when the heater was supplied with stand pipe water. From Chart No. 1 was selected the maximum "K" as shown for each back pressure, and which was the maximum obtainable under the actual conditions of the test. This value of "K" was then substituted in a formula which was derived from that given above and which should show the maximum capacity at the specified temperatures of the water and steam, but at the test back pressure. This formula is as follows:—
Capacity in pounds per hour =

$$\left(\frac{K \times 51.3}{1.64 \times 11.8 \times 150.2}\right)^{2.5}$$

(Assuming mean temperature of steam maintained at 218.5 degrees F.)

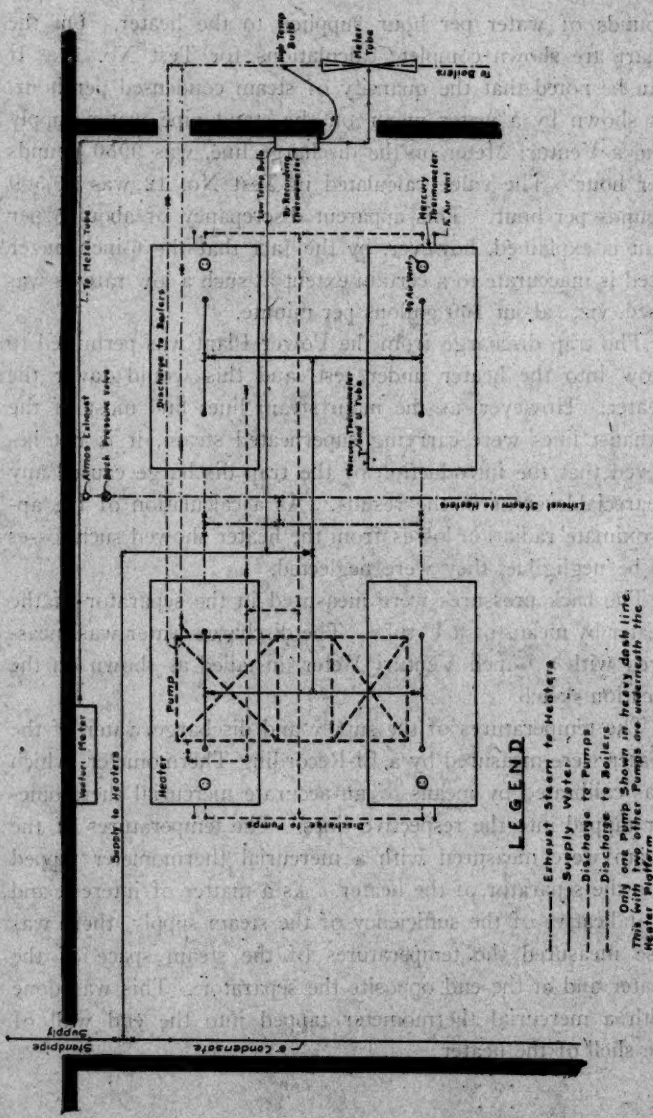
On Chart No. 2, these calculated values of the capacity were plotted against the respective back pressure. By extrapolation the curve which had been produced would indicate (under specified conditions) a maximum capacity of 85,000

pounds of water per hour supplied to the heater. On the chart are shown complete calculations for Test No. 12. It can be noted that the quantity of steam condensed per hour, as shown by a water meter on the stand pipe water supply and a Venturi Meter on the discharge line, was 9950 pounds per hour. The value calculated in Test No. 12 was 10,500 pounds per hour. This apparent discrepancy of about 5 per cent is explained, however, by the fact that the 6-inch meter used is inaccurate to a certain extent at such a low rate as was used, viz.: about 160 gallons per minute.

The trap discharge from the Power Plant was permitted to blow into the heater under test, and this would favor the heater. However, as the main steam lines and most of the exhaust lines were carrying superheated steam, it is not believed that the introduction of the trap discharge caused any appreciable error in the results. As a calculation of the approximate radiation losses from the heater showed such losses to be negligible, they were neglected.

The back pressures were measured in the separator of the heater by means of a U tube. The discharge water was measured with a 6-inch Venturi Meter, installed as shown on the location sketch.

The temperatures of the supply and discharge water of the heater were measured by a Bi-Recording Thermometer, which was calibrated by means of an accurate mercurial thermometer tapped into the respective lines. The temperatures of the steam were measured with a mercurial thermometer tapped into the separator of the heater. As a matter of interest, and as indicative of the sufficiency of the steam supply, there was also measured the temperatures of the steam space of the heater and at the end opposite the separator. This was done with a mercurial thermometer tapped into the end wall of the shell of the heater.


To Boilers
Heater Tube
Low Temp Bulb
To Recording Thermometer
Mercury Thermometer & Air Vent
To Heater Tube
Discharge to Boilers
To Air Vent
Atmos Exhaust
Back Pressure valve
Mercury Thermometer Inlet & Tube
Exhaust Steam to Heaters
Water Meter
Supply to Heaters
Pump
Heater
Discharge to Pumps
Standpipe Supply
To Condenser
LEGEND
Exhaust Steam to Heaters
Supply Water
Discharge to Pumps
Discharge to Boilers
Only one Pump Shown in heavy dash line. This with two other Pumps are underneath the Heater Platform

In the early days of the operation of the heater, it was found that the air valves, placed on the top of the heater, did not properly vent the heater when operating under low back pressure. In fact the heaters became air bound several times, and the feed pumps lost water in consequence. A notched pipe was then run through the trap discharge tapping of the heater, and carried up about a foot above the top of the heater. This pipe is half inch in diameter, has notches at the bottom, and reaches the entire length of the interior of the heater. Such method of venting obviated any further trouble with air.

The conditions under which the various tests were made were as follows:—

Test No. 1. As found—back pressure valve wide open—air valve open and inoperative—open vent through notched pipe—one heater in service—condensate supply.

Test No. 2. As test No. 1.

Test No. 3. Back pressure valve closed 3½ turns—air valves removed and tappings plugged—condensate supply.

Test No. 4. Same as 3, except stand pipe water supply.

Test No. 5. Same as 4, except two new heaters in service. All readings taken as one heater.

Test No. 6. Air valve tappings plugged—open vent through notched pipe—one heater in service—stand pipe water supply.

Test No. 7, 8, 9, 10, 11, 12, 13 and 14. Same as No. 6.

Test No. 15. Vent through one air valve tapping (⅜ inch)—notched pipe vent closed—one heater in service—stand pipe water supply.

Test No. 16. Same as 15.

Test No. 17. Air valve tapping plugged—open vent through notched pipe—one heater in service—stand pipe water supply.

Test No. 18. Same as 17.

The shortest test was of 70 minutes duration, and the longest test was 125 minutes. Observations of all quantities were taken every 5 minutes. Tests 1 to 5, inclusive, are chiefly of interest in showing the effect of using condensate supply, which has little air in it, as against using stand pipe water which has considerable air in it. Tests 15 and 16 are of interest in that a slightly increased capacity is indicated for the use of a ⅜-inch unrestricted vent at the top of the heater. It was found that at the higher rates and pressures, the ½-inch notched pipe through the trap discharge tapping was more or less obstructed with water. However, the free venting through the air valve tapping at the top of the heater is very undesirable from the standpoint of economy and comfort. About 90 pounds of steam per hour would be discharged at the higher back pressures. This would equal about 2/3 of 1 per cent of the steam utilized in the heater. On the other hand the loss of steam through the notched pipe is noticeably less, and frequently a cool discharge of air can be felt.

Various corrections were made to the measuring instruments. Corrections to the low temperature reading of the Bi-Recording Thermometer ranged from 5 degrees to 8 degrees F., and corrections to the high temperature reading ranged from — 4.2 to + 2.8 degrees F.

The results of the tests are not as accurate, of course, as the results obtained in a laboratory for two or three reasons.

(*a*) Use of superheated steam introduced a slight error.

(*b*) Use of the total pressure of steam and air present in the heater, instead of the partial pressure of the steam, introduces a slight error.

MEAN DATA OF TESTS MADE UPON A "COCHRANE" OPEN FEED WATER HEATER,

INSTALLED AT THE U.S. NAVAL PROVING GROUND INDIAN HEAD MD. UNDER CONTRACT Y-W.D.3924.

Test No.	Date 1921	Rate of Water Discharged as indicated by Venturi Meter (lbs/hr)	Pressure in Separator (lbs/sq.inch)	Pressure of Air Vent (in Inches)	Barometer (Millimeters) (°C)	Temperature in Separator (°F)	Temp. of Steam Space of Separator (°F)	Temp. of Water in Heater (°F)	Temp. of Water Supply (°F)	Remarks
1	Oct 3	68,900	0.05		764.0-21.5	228.0		96.5	70.52	Condensate Supply; All Vents Open
2	" 6	67,400	0.03	0.587	760.0-21.0	224.3	209.0	99.7	199.1	As Test #1
3	" 6	74,400	0.06	0.986	761.5-21.1	234.2	209.5	97.0	204.7	Condensate Supply, Air Valve Stopping a Pressure; other Vents Open.
4	" 7	59,300	0.09	0.426	760.1-21.4	220.4	208.3	65.8	212.3	As Test #3, except Standpipe Water Supply.
5	" 7	72,700	0.03	0.348	760.1-21.4	228.8	204.3	65.0	176.0	As Test #4, except Two Heaters in Service.
6	" 12	79,300	0.46		760.0-21.0	240.3	206.4	62.8	206.4	Standpipe Water Supply, Air Valve Stopping Pressure, Other Vents open, One Heater in Service.
7	" 18	81,100	1.21		761.0-21.4	279.4	214.7	72.5	211.0	do
8	" 19	101,400	0.81		754.4-21.3	268.6	211.7	67.9	203.2	do
9	" 19	106,200	1.21		754.4-21.3	282.9	211.9	69.0	203.0	do
10	" 22	37,400	0.44		760.0-21.1	248.5	211.2	65.2	206.3	do
11	" 25	80,000	0.27		760.0-21.0	265.2	211.2	69.1	203.0	do
12	" 26	91,400	0.26		769.5-21.2	288.6	211.2	62.7	197.1	do.
13	" 28	101,000	0.24		767.0-21.0	269.4	210.2	66.0	188.3	do
14	" 29	102,600	0.75		769.5-21.0	281.5	212.3	68.0	205.2	do.
15	" 29	104,200	0.79		767.5-21.0	281.5	212.6	68.5	205.7	As Test #14 except Vent only thru Air Valve Stopping.
16	" 29	106,900	1.13		766.2-21.1	284.3	213.6	69.8	207.4	As Test #15
17	Nov 5	91,300	0.80		763.0-21.0	274.8	211.0	59.6	206.9	As "14 except Air Valve Stopping Pressure, other Vents Open.
18	" 5	102,000	1.20		763.0-21.0	278.4	213.4	60.2	208.2	As Test #17

CALCULATED RESULTS OF TESTS MA

INSTALLED AT THE U.S. NAVAL PROV

Test No.	Date 1921	Actual Rate of Water Discharged (lbs/hr)	Steam Condensed (lbs/hr)	Water Supplied to Heater (lbs/hr)
1	Oct 3	67,600	6750	60,800
2	" 6	66,200	6,620	59,600
3	" 6	73,000	7,170	65,900
4	" 7	58,300	6,520	51,900
5	" 7	71,700	7,090	64,600
6	" 12	77,800	9,930	67,900
7	" 18	78,500	9,670	69,700
8	" 19	99,300	11,900	87,800
9	" 19	104,300	12,220	92,100
10	" 22	36,700	4,600	32,100
11	" 25	78,500	9,350	69,100
12	" 26	89,900	10,300	79,300
13	" 28	99,400	10,600	88,800
14	" 29	100,600	12,340	88,300
15	" 29	102,200	12,490	89,700
16	" 29	104,800	12,910	91,900
17	Nov 5	89,600	11,000	77,000
18	" 5	100,000	13,200	86,800

TEST #12 ON "COCHRANE" OPEN FEED WATER HEATER AT INDIAN HEAD MD

IN OPERATION:- 3000 K.W. Generator
#3 Vac Pumpt New Feed Pump
#3 and #4 Pans
2 Stoker Engines

OCT. 26 1921

BAROMETER 769.5-21.2°C

STEAM CONDENSED; AS INDICATED

WATER SUPPLY & VENTURI M

LOG SHEET

Time Oct 26 1921 A.M.	Rate of Discharge (lbs/hr)	Venturi Meter Reading (lbs)	Pressure in Sepr. (in. Hg)	Tempr. in Sepr. (°F)	Temperature of Discharge Water (°F)	Temp. of Steam Space of End Opposite to Sepr. (°F)	Temperature of Supply Water (°F)	Load (K.W)	Vacuum (in. Hg)	Water Meter Reading (gal)
10:40	96,000	502,684,600	1/8	298	200	211	66	1700	28.3	4,685,240
45	90,000	692,900	1/2	295	197	211	67	1650	28.3	6,980
50	78,000	699,900	1/8	294	196	211	68	1700	28.3	6,880
55	86,000	707,500	1/8	294	197	211	68	1700	28.3	7,640
11:00	88,000	714,300	1/2	293	199	211	68	1800	28.3	6,370
05	87,000	721,300	1/2	292	200	211	70	1600	28.3	7,180
10	102,000	728,700	1/2	291	199	211	71	1650	28.3	7,340
15	94,000	736,800	1/2	297	198	212	70	1750	28.2	4,690,940
20	86,000	744,100	7/8	283	198	212	71	1700	28.2	1,660
25	90,000	752,000	1/2	283	196	212	71	1750	28.2	2,430
30	92,000	759,600	1/2	283	198	211	69	1750	28.2	3,220
35	94,000	768,000	1/2	284	199	211	70	1790	28.1	4,050
40	94,000	775,800	1/2	286	198	211	71	1800	28.1	4,910
45	93,000	783,800	1/2	286	197	211	70	1700	28.1	5,740
50	93,000	791,800	1/2	287	197	211	70	1700	28.1	6,600
55	93,000	799,700	1/2	288	197	211	71	1725	28.1	7,440
12000	95,000	807,800	1/2	290	197	211	72	1800	28.1	8,30
05	96,000	816,000	1/2	286	195	211	72	1700	28.1	9,100
Totals	1645,000		25.5/8	5198	3548	3801	1255			
Means	91,400		0.25/8	288.8	197.1	211.2	69.7			

Venturi Meter Readings	Diff. (lbs) (5 min reading)	Tempr. (°F)	Corrected Value of Diff. (lbs)
502,684,600		200	
692,900	83 00	197	8
699,900	7 000	196	6
707,500	7,600	197	7
714,300	6,800	199	6
721,300	7,000	200	6
728,700	7,400	199	7
736,800	8,100	198	8
744,100	7,300	198	7
752,000	7,900	196	7
759,600	7,600	198	7
768,000	8,400	199	8
775,800	7,800	199	7
783,800	8,000	197	7
791,800	8,000	197	7
799,700	7,900	197	7
807,800	8,100	197	8
816,000	8,200	198	8
Totals			12
Mean			

TS OF TESTS MADE UPON A "COCHRANE" OPEN FEED WATER HEATER

& NAVAL PROVING GROUND INDIAN HEAD MD UNDER CONTRACT WD 3824

Condensed (lbs/hr)	Water Supplied to Heater (lbs/hr)	Heat Supplied to Water (Btu/hr)	Mean Temp. of Steam (°F)	Logarithmic Mean Temp. Difference (°F)	Coefficient of Heat Transfer	Remarks
6750	60,800	6,620,000	212.4	39.0	4200	Condensate Supply: All Vents Open.
6,620	52,600	6,520,000	212.1	40.8	3500	As Test #1. Condensate Supply: All Valves Throttled Slightly.
7,170	65,900	7,884,000	212.3	39.0	4300	
6,520	51,000	6,460,000	212.3	65.1	2950	As #2 except Standpipe W. Supply
7,090	84,600	7,168,000	212.3	79.3	2400	As #4 except Two Heaters in Service
9,930	67,900	9,770,600	213.7	48.0	4900	Standpipe Water Supply. One Heater in Service.
9,670	69,700	9,850,000	216.8	44.0	5200	do.
11,900	87,800	11,900,000	215.3	53.9	4670	do.
12,220	92,100	12,350,000	216.8	56.5	4910	do.
4,600	32,100	4,540,000	214.0	47.0	3600	do.
9,350	69,100	9,341,000	213.4	48.0	4600	do.
10,300	78,300	10,700,000	214.7	62.2	3940	do.
10,600	88,600	10,800,000	214.0	69.2	3390	do.
12,340	98,300	12,110,000	215.1	51.8	4680	do.
12,490	89,700	12,300,000	215.2	58.6	5000	As #10 except Vent only Three Air Valves Tapping.
12,910	91,900	12,700,000	216.4	59.0	5100	As #15
11,800	77,800	11,500,000	214.9	49.0	5200	As #16 except All Vents Valves Throttled Other Vent Open.
13,200	86,800	12,800,000	216.2	49.6	5300	As #17

COMPUTATIONS of T

$\frac{14.88}{.28}$ / $\frac{216-15.28}{213-14.88} = \frac{13.16}{4.12}$

15.16% / $\frac{13}{19}$

15.16 % = 213.51°F / 13/100 X 10 = 0.51°F

Total Heat @ 15.16 % 214·9640+182.3=1147.2

= 1146.9+0.17+9.11 213·9656+181.3=1138.9

= 1147.0+35 = 1182.0 .37 X 3 = 0.17

208.6-213.6 = 7.5

75 X 46 = 33 B.t.u.

Correction for Water @ 197.1°F = 91,400 X 1.5%00

= 91,400 - 1600 = 89,800 #/hr

89,800 = x(1632 - 30.9) = 2(1101 - 185.2)

(89,800 - 2)1144 = 1917 or 12,100,000 = 1151X

X = 10,500 #Steam Condensed/hr.

10,500 X 1917 = 10,700000 Btu/hr Supplied to Water.

Logarithmic Mean Temp. Difference

$d = \frac{t_2 - t_1}{\log \frac{T_2 - t_1}{T_1 - t_2}}$ $\frac{197.1}{12.5}$ $\frac{213.6}{185.2}$ $\frac{28.4}{18.6}$

= 15.7 = Mean Temp of Water 99.7 121.2

Then, Mean Temp. of Steam = $\frac{(147.1)(28.4)+(28.6)(121.2)}{1183}$ + 129.9

= $\frac{2400+4200}{1183}$ +129.9 = 84.8 + 129.9 = 214.7°F

Then, $d = \frac{121.1-61.7}{\log \frac{121.1-99.1}{61.7-99.1}} = \frac{1246}{\log 1.58}$ = $\frac{1246}{2.3026 \times 0.1987}$ = 62.2°F

Transmission Efficiency

$10,700,000 = K\left(\frac{32,000-10,700}{62.2}\right)^{0.6}\left(\frac{62.2}{62.2-32}\right) = K\left(1290^{0.6}\left(\frac{62.2}{61.2}\right)\right)$

$= K(73.5)\frac{62.2}{30.8}$ $\therefore K = 3,940$

Approximate Radiation Losses

2 X 9.75 X 5.42 = 94.5

2 X 2.42 X 8.00 = 38.6

2 X 9.75 X 9.00 = 14.00

327.1 #

327(212 - 70).30 = 325 A 142 A.30 = 14,000 B.T.U./hr

100% X $\frac{14000}{12,800,000}$ = 0.11% of total heat supplied in Test No 18

NDIAN HEAD MD

: AS INDICATED BY WATER METER ON THE STANDPIPE

& VENTURI METER ON THE DISCHARGE (IN lbs/hr)

(lbs) reading	Temp: (°F)	Corrected Value of Differences (lbs/smin)	Water Meter Readings (U.S. gals)	Differences (gals/5min)	Differences (lbs/5min)	Steam Condensed (lbs/5min)
	200		4,685,240			
00	197	8,200	6,080	840	7020	12 00
00	196	6,900	6,880	840	6,680	2 00
00	197	7,500	7,640	760	6,340	1 2 00
00	199	6,700	8,370	730	6,090	6 00
00	200	8,900	9,180	810	6,760	1 00
00	199	7,300	9,940	760	6,340	1 0 00
00	193	8,000	4,690,940	900	7,510	5 00
00	193	7,200	1,680	820	6,840	4 00
00	196	7,800	2,430	770	6,430	1 4 00
00	198	7,500	3,220	730	6,090	9 00
00	199	8,300	4,050	830	6,930	1 4 00
00	198	7,700	4,910	860	7,180	5 00
00	197	7,900	5,740	830	6,930	1 0 00
00	197	7,980	6,600	860	7180	7 00
00	197	7,900	7,440	840	7010	8 00
00	197	8,000	8,300	860	7,180	8 00
00	195	8,100	9,100	800	6670	1 4 00
		129780			115,690	14,100
						829

' OPEN FEED WATER HEATER.

HEAD MD. UNDER CONTRACT Y.W.D. 3924.

Temp. of Supply water (°F)	Remarks
052.	Condensate Supply. All Vents Open.
99.1	As Test #1
04.7	Condensate Supply. Air Valve Trappings Plugged: other Vent Open.
12.3	As Test #3, except Standpipe Water Supply.
76.0	As Test #4, except Two Heaters in Service.
06.4	Standpipe Water Supply. Air Valve Trappings Plugged. Other Vent open. One Heater in Service.
11.0	do
03.2	do
03.0	do
06.3	do
03.8	do
97.1	do
88.3	do
05.2	do
05.7	As #10 except Vent only One Air Valve Trappings
07.4	As Test #15
06.9	As #18 except Air Valve Trappings Plugged; other Vent Open.
08.2	As Test #17

CALCULATED RESULTS OF TESTS MADE UPON A "COCHRANE" OPEN FEED WATER HEATER

INSTALLED AT THE U.S. NAVAL PROVING GROUND, INDIAN HEAD MD. UNDER CONTRACT Y/W.D. 3924.

Test No.	Date 1921	Actual Rate of Water Discharged (lbs/hr)	Steam Condensed (lbs./hr)	Water Supplied to Heater (lbs./hr)	Heat Supplied to Water (B.t.u.)	Mean Temp. of Steam (°F)	Logarithmic Mean Temp. Difference (°F)	Coefficient of Heat Transfer	Remarks
1	Oct 3	67,600	6750	60,800	6,620,000	212.4	39.0	4280	Condensate Supply. All Vents Open.
2	" 6	66,200	6,620	59,600	6,520,000	212.1	40.8	3580	As Test #1.
3	" 6	73,000	7,170	65,800	7,064,000	212.3	39.0	4300	Condensate Supply. All Valve Trappings Plugged.
4	" 7	58,300	5,520	51,800	6,460,000	212.3	65.1	2950	As #3, except Standpipe W. Supply
5	" 7	71,700	7,090	64,600	7,164,000	212.3	79.2	2400	As #4 except Two Heaters in Service.
6	" 12	77,900	9,930	67,900	9,770,600	213.7	48.0	4900	Standpipe Water Supply. One Heater in Service.
7	" 18	78,500	9,670	69,700	9,830,000	216.8	44.0	5200	do.
8	" 19	98,500	11,900	87,800	11,900,000	215.3	53.9	4670	do.
9	" 19	104,300	12,220	92,100	12,350,000	216.8	56.5	4510	do
10	" 22	36,700	4,600	32,100	4,540,000	214.0	47.0	3600	do.
11	" 25	78,500	9,350	69,100	9,361,000	213.4	48.0	4600	do.
12	" 26	89,900	10,200	79,300	10,700,000	214.7	62.2	3940	do.
13	" 28	99,400	10,600	88,800	10,800,000	214.0	69.2	3390	do.
14	" 29	100,600	12,340	88,300	12,118,000	215.1	51.0	4880	do.
15	" 29	102,200	12,490	89,700	12,300,000	215.2	50.0	5000	As #14, except Vent only thru. Air Valve Trapping.
16	" 29	104,800	12,910	91,900	12,700,000	216.4	50.0	5100	As #15
17	Nov 5	89,600	11,800	77,800	11,500,000	214.9	49.0	5200	As #16, except All Valve Trapping Plugged, Other Vent Open.
18	" 5	100,000	13,200	86,800	12,800,000	216.2	49.0	5300	As #17

TEST #12 ON "COCHRANE" OPEN FEED WATER HEATER, AT INDIAN HEAD MD.

Pump — OCT. 26, 1921

BAROMETER: 762.5 @ 12°C

STEAM CONDENSED AS INDICATED BY WATER METER ON THE STANDPIPE WATER SUPPLY, & VENTURI METER ON THE DISCHARGE (IN lbs./hr.)

Temp. (°F)	Temperature of Supply Water (°F)	Load (K.W)	Vacuum (in. Hg)	Water Meter Reading (U.S. gals)	Venturi Meter Readings	Diff. (lbs) (5 min. reading)	Tempr. (°F)	Corrected Value of Differences (lbs./5 min)	Water Meter Readings (U.S. gals)	Differences (gals./5 min)	Differences (lbs./5 min)	Steam Condensed (lbs./5 min)
	66	1700	28.3	4695,240	302,684,600		200		4,695,240			
	67	1650	28.3	6,090	692,900	8300	197	8,200	6,090	840	7020	1200
	68	1700	28.3	6880	699,900	7000	196	6,900	6,880	840	6690	200
	68	1700	28.3	7,940	707,500	7,600	197	7,500	7,640	760	6340	1200
	68	1900	28.3	8,370	714,300	6,800	199	6,700	8,370	730	6090	600
	70	1600	28.3	9,180	721,300	7,000	200	6,900	9,180	810	6760	100
	71	1650	28.3	9,940	728,700	7,400	199	7,300	9,940	760	6340	1000
	70	1750	28.2	4,690,840	736,800	8,100	193	8,000	4,690,940	900	7510	500
	71	1700	28.2	1,660	744,100	7,300	193	7,200	1,680	820	6840	400
	71	1750	28.2	2,430	752,000	7,900	196	7,800	2,430	770	6430	1400
	69	1750	28.2	3,220	759,600	7,600	198	7,500	3,220	790	6590	900
	70	1750	28.1	4,050	768,000	8,400	199	8,300	4,050	830	6,930	1400
	71	1800	28.1	4,910	775,800	7,800	198	7,700	4,910	860	7,180	500
	70	1700	28.1	5,740	783,800	8,000	197	7,900	5,740	830	6,930	1000
	70	1700	28.1	6,600	791,800	8,000	197	7,900	6,600	860	7180	700
	71	1725	28.1	7,440	799,700	7,900	197	7,800	7,440	840	7010	800
	72	1800	28.1	8,30	807,800	8,100	197	8,000	8,300	860	7180	800
	72	1700	28.1	9,100	816,000	8,200	195	8,100	9,100	800	6670	1400
	1255				Totals-			129760			115,680	14,100
	69.7				Mean							829

HEATER

ACT WB 3824.

COMPUTATIONS of TEST # 12

BAROMETER: 762.8 @ 12°C

marks	
Supply:	
to:	
of 1.	
ngth: All	
Piping.	
ndpipe W. Supply	
e Nozzles in Service	
ter Supply	
in Service.	

(*c*) Introduction of trap discharge introduces a slight error, etc.

If the usual method of testing, including weighing of water, was used, greater accuracy could be secured. But it is believed that the method used is simple, can be used in any Power Plant, with but slight trouble, is easily adaptable to any type of heater, and is sufficiently accurate for all practical purposes.

THE PRODUCTION OF THE MINES FOR THE NORTHERN MINE BARRAGE.

By Commander F. J. Cleary, U. S. Navy, Member.

The actual laying of the Northern Mine Barrage has been celebrated in song and story, but the manufacture and delivery of the mines to the mine laying force—an achievement equally remarkable in conception and execution—has never been presented to the public. Its accomplishment was due to standardization, not only of parts and methods of manufacturing and assembly, but also of ships and of men.

The proposal to construct a mine barrage 250 miles long in water reaching the depth of 980 feet and make the whole stretch of 250 miles exceedingly dangerous from the surface to a depth of 300 feet was unprecedented, and up to that time had been considered impossible from every practical standpoint, and it was impossible with the mines then in use. Time was a vital factor, as every day saw the loss of many priceless ships and cargoes.

From April, 1917, to July, 1917, the Navy made an intensive study of all types of barrages; the three types promising any success were:

(1) Nets and entanglements.

(2) Nets combined with mines.

(3) Mines alone.

The war experience of the British proved the unsuitability of barrages of types (1) and (2), and decision was then made that the use of mines alone offered the only practicable solution. At least 100,000 mines would be required, and to be of

any real value they would have to be of a type more suitable for anti-submarine operations than any mine then in existence. In other words, a successful anti-submarine mine had first to be developed and then manufactured at a rate far in excess of any production that had ever been reached even under the stress of war.

The novel principle of the firing mechanism of this mine which made it successful and which required the use of only one-third as many mines as would be required by the best of the old type was discovered in April, 1917, but was not brought to a state of development warranting its adoption until late in July, 1917, and the entire mine was designed around the firing mechanism. The possibility of the northern barrage depended absolutely upon the success of the new mine. Had nothing better than the then existing types of mine been available, the northern barrage would have been utterly impossible of execution within the time available by reason of the enormous number of mines required. The combined resources of the United States and the Allies, especially in the production of high explosives, could not have produced the required number of mines nor could the combined mine laying forces have planted them in a single year.

On August 15, 1917, Admiral Mayo, U.S.N., who was about to proceed to England, obtained full details of the proposed mine barrage to present to the British Admiralty. Immediately upon his return a conference was held on October 15, 1917; the suggestions of the British Admiralty were discussed and adopted and the Bureau of Ordnance directed to procure 100,000 mines, the other Bureaus of the Navy Department undertaking all other necessary preparations. The planting of the mine barrage was to begin as early in the spring of 1918 as possible to assure its completion during the favorable weather of summer and early fall. Therefore only four months were available in which to complete the details of de-

sign and start manufacturing on an extensive scale so as to obtain production in adequate quantities not later than February, which was necessary in order that the mines could be shipped abroad, assembled and made ready for laying by April, 1918. This program necessitated their manufacture at the rate of 1000 per day or 4.2 times the production the British had succeeded in reaching after three years of war.

In addition to manufacturing the mines themselves the other features for the entire operation were as follows:

1. Building a mine loading plant and obtaining a shipping point.

2. Shipment of raw material and finished parts to the various manufacturers and to the loading and shipping points.

3. Securing cargo tonnage to transport mines and mining material overseas.

4. Conversion of merchant ships into mine layers.

5. Training officers and men for the loading plant and shipping base, assembly bases overseas, and crews for mine layers.

All of this work, involving the handling of a tremendous amount of material, was but one of the many major naval operations then in hand, and every minute detail had to be followed up continuously to prevent overlooking some essential part, the delay of which would hold up the whole project.

On November 1, 1917, the only parts of the mine that had been completely designed were the firing mechanism and the mine case. Usually a new mechanism as complicated as a mine can hardly be expected to function as designed until complete models have been made and tested under service conditions and the usual minor defects discovered and remedied. Ordinarily it requires at least a year to prove out such a mechanism before placing it in quantity production. A new mine based on an entirely new principle had to be built, it had to be efficient, and yet had to be capable of manufacture and assembly in great quantities in the least possible time.

Anticipating the adoption of the mine barrage project, the Bureau of Ordnance had placed an order for 10,000 firing mechanisms on August 9, 1917, and for 90,000 additional mechanisms on October 3, 1917.

There were no commercial plants in the United States that had had any experience in mine manufacture.

It was decided that the only method of manufacture which would result in the early quantity production desired was to utilize the wonderful resources of the passenger automobile factories by designing the mine as an assembly proposition, by standardizing all parts, and by authorizing the use of commercial processes and standards wherever possible. By the use of this method the many parts of the mine could be manufactured in different factories with great rapidity and could be finally brought together and assembled without any cutting and fitting.

The complete mine was therefore divided into several groups, each group being a separate design and manufacturing problem, all so standardized that the several groups would be interchangeable and would assemble into a complete mine ready for the final adjustments and use. The several groups were:

Firing mechanism.
Extender mechanism.
Mine case.
Anchor.
Antenna and floats.
Horn device.
Release gear.

Each one of these groups was composed of many parts, the parts of the firing mechanism alone being manufactured by more than twenty different factories, the complete assembled mine being composed of several hundred parts. Each group

was manufactured and tested quite independently of the other groups.

The mines complete were manufactured by 140 contractors and over 400 subcontractors; the mine cases were produced by 4 steel fabricating plants; the mine anchors, composed of many different parts, were manufactured by 3 automobile firms; the wire rope by 10 manufacturers, the firing mechanisms by 2 electrical manufacturers. Dimensioned drawings were furnished each contractor and subcontractor. The drawing tolerances specified for fitted parts were of the two following classes:

Extreme accuracy required .002 inch to .005 inch.
Fair accuracy required .010 inch to .015 inch.

The manufacturers made their own gauges to assure these tolerances.

It was an invariable rule to divide the order for any one part between at least two contractors or subcontractors so that the failure of any one manufacturer would only slightly delay production. A production of 1000 mines per day was planned, and it was found after getting into production that this rate could easily be exceeded if desired.

As an indication of the vast amount of material involved in manufacturing 100,000 mines, it is of interest to note that the dead weight of the mine anchors alone exceeded 40,000 tons, that over 19,000 miles of 7/16 inch wire rope were used for mooring the mines, that 30,000,000 pounds of TNT was used, and that the mine cases required approximately 3,500,000 square feet (or 7800 tons) of steel plate ⅛ inch thick.

Complete mines were not available for test until March, 1918, and with the exception of a very few mechanical faults which were readily corrected, the mine and anchor functioned as designed—a remarkable tribute to the small group of offi-

cers of the Navy who designed the mine and planned and carried through its manufacture, and to the great number of men engaged in their manufacture.

This mine was designed especially for safety in handling. That this safety was attained is shown by the facts that 85,000 of these mines were loaded and shipped overseas and assembled and 57,000 of them planted in the barrage without a single accident to the thousands of naval personnel handling them.

A vital reason for the method of manufacture adopted was that only by this means could secrecy be preserved regarding the characteristics of the mine. It is obvious that if 100 different parts are manufactured in 100 different factories that are in ignorance of the fact that they are manufacturing mine material, no individual could obtain sufficient information to visualize the complete mine and thereby disclose its secret to the enemy. It was felt that if the mines could be manufactured and planted before the enemy had discovered the secret of the novel firing mechanism the war would be won for the Allies before he could develop a sufficient protection against it. Consequently the vast majority of factories manufacturing the parts did not know that they were intended for mines. The one company manufacturing the firing device took such precautions that only three officials of the company knew that the electrical apparatus being manufactured was intended for use in a mine. This secrecy was carried still farther to the extent that even at the points where material was assembled for shipment overseas, the parts were not assembled into the complete mine, but were shipped in groups to the overseas assembly bases. Excepting a few mines assembled aboard ships of the mine force for testing purposes, no mines were completely assembled in this country, and in spite of the German secret service and many sympathizers in the United States, the secret of the new mine was kept from the enemy until long after the barrage was planted. Another advantage of

shipping the mines disassembled was in conserving cargo space, as an assembled mine occupied approximately three times as much space as the parts of a disassembled mine. Although this method of shipping disassembled required large assembly bases overseas, this complication was more than offset by the increased shipping capacity gained.

The extreme severity of the winter of 1917-1918, the fuel shortage, the congestion of traffic, the railroad embargoes and labor troubles all operated to delay production and for weeks the situation was critical. It is estimated that these handicaps delayed the beginning of quantity production about six weeks in spite of using every expedient such as motor truck transportation, shipment of whole trainloads of material on passenger schedules, etc.

As one item of the barrage project it became necessary to create a complete mine loading plant capable of receiving, loading and shipping 1000 mine cases per day, there being no plant in the United States capable of handling this amount of material. This plant was started in October, 1917, and completed in March, 1918. It covered an area of 3000 feet × 800 feet and consisted of 22 buildings, including a mine case storage building capable of storing 5000 empty mine cases, a melting plant capable of melting and pouring a minimum of 300,000 pounds of TNT per day, a cooling building capable of holding 1,000 loaded mine cases, a ready storage building with a capacity of 4,000,000 pounds of TNT. The entire plant was equipped with conveyors and other labor saving facilities. The rated capacity of 1000 mines per day was exceeded by 50 per cent on one occasion, and a total of more than 73,000 mines, involving the melting and handling of over 22,000,000 pounds of TNT were loaded here without accident. Loaded mine cases were shipped direct from this plant to the bases overseas. The personnel of the plant consisted of about 400 enlisted men of the Navy and from 200 to 400

additional enlisted men were employed in loading the mine cases into the cargo vessels. In order to prevent any delay in shipment of loaded mine cases overseas that might result from delay in completing this plant, an order was placed with the Dupont Company to load 17,000 mine cases with TNT at its Wisconsin plant. These were loaded in February and March, 1918, delivered at the naval mine loading plant, and from there shipped overseas.

As the mines were to be assembled at the overseas bases, it was necessary to maintain an adequate supply of all the component parts at those bases, requiring a steady flow of parts from all factories to the shipping point at Norfolk and thence overseas. Adequate shipping facilities at Norfolk were secured by taking over the S.P.R.R. Pier, No. 4. This pier was 875 feet × 270 feet and was able to accommodate seven cargo vessels at a time. All mine material, except the loaded mine cases, was shipped from this point. This pier was used for storage as well as for shipping; it had capacity for 40,000 mine anchors and other non-explosive parts. A force of about 400 enlisted men of the Navy was constantly employed at this pier.

In the inception of the mine barrage project, the Navy had secured from the Shipping Board 24 cargo vessels of the *Lake* class for exclusive use to carry mine material from Norfolk to the overseas bases. These vessels were armed with naval guns, manned by naval crews and handled by the Navy. They were of 3000 tons dead weight capacity, and a typical cargo for each ship was made up of 2000 mines with anchors and fittings complete, 500 tons of miscellaneous naval supplies, about 500 tons of fuel oil for naval vessels overseas, and about 500 tons of extra bunker coal. The sailings of these carriers averaged two ships every 7 or 8 days, half in Norfolk convoys and half in Halifax convoys. It took a ship in convoy about twenty days from loading point to discharging point

and from 65 to 70 days were required for a round trip. Of the 24 carriers only one, the *Lake Moor*, was lost, sunk by an enemy submarine off the Irish coast on April 11, 1918, with the loss of most of her crew and about 1500 tons of mine material.

When the United States entered the war, the mine force of the Navy consisted of two mine layers, the *San Francisco* and the *Baltimore*, with a combined carrying capacity of 350 mines. On the basis of an estimated output of 5000 mines per week and of one mine laying operation per week, it was necessary to expand the mine laying force to a capacity of at least 5000 mines, which, if all went well, would insure the completion of the northern barrage in three months. This expansion involved converting the requisite number of merchant ships and training officers and crews for them.

Working in co-operation with the Shipping Board the Navy took over the following eight ships:

Old Name	*New Name*	*Date Taken Over*	*Converting Yard*	*Mine Capacity*
El Dia	*Roanoke*	Nov. 16, 1917	Tietjen & Lang	850
El Rio	*Housatonic*	Nov. 25, 1917	Tietjen & Lang	850
El Siglo	*Canandaigua*	Nov. 22, 1917	Morse Shipyard	850
El Cid	*Canonicus*	Nov. 24, 1917	Morse Shipyard	850
Massachusetts	*Shawmut*	Nov. 6, 1917	Boston Navy Yard	300
Bunker Hill	*Aroostook*	Nov. 10, 1917	Boston Navy Yard	300
Jefferson	*Quinnebang*	Dec. 2, 1917	Robbins Shipyard	600
Hamilton	*Saranac*	Dec. 6, 1917	Shewan & Sons	600
				5200
San Francisco and Baltimore				350
				5550

The conversion of these eight ships was a work of considerable magnitude. It involved clearing the greater part of the second and third decks for carrying mines; adding water tight bulkheads; clearing away parts of upper deck structures; providing quarters for additional officers; closing cargo ports; providing chutes for coaling when mines were aboard; cutting

two or four ports in the stern for launching mines; installing mine elevators; repairing and strengthening decks; building gun platforms, lookout stations, navigating and signal bridges; altering boat stowage and davits for carrying heavier boats; installing davits, booms and winches on upper deck for embarking mines; providing messing and berthing arrangements for a crew of about 400 men; installing bakery and galley; building storerooms, magazines, fresh water tanks, additional coal bunkers, machine shop and elevator machinery rooms; installing fire system, magazine sprinkler system, steam heating system, speaking tubes and telephones for gun fire control and other communications; overhauling main machinery; replacing anchor engine and windlass with one of more powerful type to handle the heavier anchors and chains; adding ventilating blowers; doubling the electrical plant; adding evaporator and distiller for fresh water, hydraulic machinery for mine elevators and about ten winches on mine decks; installing machine tools and providing a larger radio telegraph plant.

The plan adopted for the mining installation of the new ships consisted of two tracks on each side of the 2d or launching deck, extending about three-fourths the length of that deck. On the deck below there were also four long tracks and additional shorter stowage tracks, where they could be fitted. Cross tracks and turntables connected all tracks at points sufficiently distributed to insure against a breakdown at any point on one track cutting off mines beyond. Mines were transferred from the lower deck to the launching deck by elevators, which was a feature unique to the United States mine layers, as elevators had been abandoned as impracticable by other navies. The elevators were of two types, electric and hydraulic, with automatic stopping and leveling devices, carried two mines and were designed to make a round trip in one minute, including the time of loading and unloading the car. This rate of speed was eventually doubled in actual practice during

mine laying. On four of the new mine layers there was room for only two mine launching ports aft and switches were installed connecting the outer track and inner track on each side. On the other four new mine layers, there was room aft for four mine launching ports, one for each track. The mines were hauled along the tracks "in fleets" of 30 to 40 mines by means of a wire rope led to a winch until they reached the launching trap when the wire rope was cast off and they were run into the trap by hand and released one at a time by a lever.

In spite of the tremendous congestion in the shipyards these eight ships were converted, officers and men assembled and trained, the ships had their trials to test out all machinery and devices and maneuvered together sufficiently so that they sailed as a squadron for the overseas bases at midnight May 11, 1918.

Two overseas assembly bases were established with a planned combined capacity of handling 3500 mines per week. These bases utilized the services of about 2500 officers and enlisted men of the Navy. The first draft of 200 men for these bases arrived in Liverpool November 27, 1917. The first supplies from the United States reached the bases January 20, 1918. The first mine carrier arrived with additional stores February 18, 1918, the first shipload of mining material arrived April 3, the second shipload April 5. On March 1 both bases had reached such a state of completion that mines could have been received and assembly work started. On April 1 the bases were ready in all respects.

The work of assembling the mines was a highly organized process developed in accordance with the standards of specialized manufacturing efficiency. The various component parts flowed from the storehouses into the assembly sheds and to each group of men handling each individual part. The two principal parts were the loaded mine case and the anchor. The

anchors moved on tracks extending across each bay of the assembly shed. Along these tracks were stationed groups of men, each group making some special adjustment. By the time the anchor reached the end of the track, the mine case, which during the same time had received its attachments and been tested while moving on a traveling table, arrived at the same point; the mine case and the anchor were joined, a few final adjustments made and the mine was ready to be placed on board a mine layer for planting. The completely assembled mines were either loaded on cars directly from the assembly sheds and run down to the wharves or they were rolled into ready issue sheds directly opposite the assembly sheds. This system proved most efficient and produced results never before attained in the rapid handling of mines.

The rate of assembling and laying mines was originally planned at 3500 per week. The assembly and possible laying of mines greatly exceeded this rate and the rate originally fixed for shipping 3500 mines per week from the United States was increased to 6000 per week. The rate of assembly at the two bases reached as high as 1340 per day. Similarly the operations of refueling, taking on water and embarking mines and supplies had been so systematized that only two days in port were necessary before the squadron was ready for the next mine laying excursion.

The sudden expansion of the mine force from 2 mine layers to 10 entailed a proportionate expansion of mining personnel. In comparison with the projected northern barrage operation, the United States Navy had had very little experience in mining, and this experience was confined to a very few officers and men. It was partly for this reason that the Bureau of Ordnance designed the mine in such a manner that it could be handled on board ship practically as "fixed ammunition" so that a minimum of experience and training of the ships' crews would be necessary to its successful use. This method

of delivering mines completely assembled, ready for planting, to the mine layers was an entirely new departure in mine laying practice and was the main reason why the mine laying squadron was able to lay such an unprecedentedly large number of mines in the northern barrage in such a short time.

On June 6, 1918, the United States mine laying squadron of ten ships left the bases fully loaded with mines and on the first excursion laid 3385 mines. The planting of the northern barrage was begun by the British in March, 1918, and up to November 11, 1918, a total of 56,760 United States mines and 16,300 British mines had been laid.

After the first two United States excursions had been completed, two enemy submarines were damaged while crossing the mined area. As the mined area grew the damages to enemy submarines increased. The *U-86* was destroyed while homeward bound; the *U.B.-22* outward bound was never heard of after she reached the barrage; the *U-113* outward bound was damaged and had to return to port; the *U-92* was sunk and on the same day another submarine outward bound was damaged and had to return to port; the *U-156* was sunk; the *U-123* was sunk. It is known beyond a doubt that 6 submarines were destroyed in the barrage and an equal number severely damaged, and the future will show in all probability that the northern barrage actually accounted for two or three times that number of enemy submarines.

The moral effect of the northern barrage was tremendous. The German submarine officers and crews dreaded mines far more than any other anti-submarine device. At the best, the submarines had but one chance in three of passing safely through the northern barrage, and the knowledge of the silent and unseen but ever present menace of the mine barrage was a grim spectre never absent from the minds of the submarine crews. The value of the northern barrage from the actual losses inflicted and from the destruction of the morale of the submarine crews is incalculable.

The number of mines laid on each excursion varied from 1596 to 5520. The record excursion for rapid laying being that of 7 September, 1918, when 5520 mines were laid in 3 hours and 50 minutes. It is interesting to note that in this 3 hours and 50 minutes there were laid more mines than the entire supply possessed by the United States when we entered the war.

As an example of what standardization and quantity production can accomplish in reducing costs, the following information will be of interest.

Prior to the beginning of the war in 1914, the United States purchased abroad 1300 mines of two different types at an average unit cost of $667.00.

In 1917 and 1918, in spite of the tremendous increased cost of labor and material over the 1914 costs, the increased cost due to overtime, Sunday and holiday work and the great increase in the cost of transporting material, the United States obtained 100,000 mines at an average unit cost of $303.77.

NOTES.

THE WHITE STAR LINER *MAJESTIC*.

In view of the general tendency of post-war shipbuilding, we imagine that several years will elapse before the claim made for the R. M. S. *Majestic* that she is the world's largest and most luxurious passenger liner is seriously challenged. By the courtesy of the managers of the White Star Line, we were given an opportunity of inspecting the vessel at Southampton on Saturday last, and although this inspection was necessarily of a somewhat cursory nature, it was fully sufficient to confirm all claims as to her appointments. The *Majestic* has a length over all of 950 feet and a breadth of 100 feet, while the height from keel to boat-deck is 102 feet; her gross tonnage is about 56,000 and her displacement 64,000 tons when loaded to her marks. There are nine decks in all, of which five are steel decks extending for the whole length of the ship, while the remainder are steel-plated erection decks covering not less than half the total length. In keeping with the main dimensions of the vessel, the public rooms are exceptionally large and lofty, and the decorations and furniture, which we cannot describe in detail, are generally elegant and comfortable. The arrangement of the rooms, it may be mentioned, has been materially facilitated by the fact that the boiler uptakes are each divided into two parts athwartships, the two parts being united above the boat deck level and connected with the funnel. The principal public rooms on the promenade deck are the lounge, palm court, restaurant and smoking room, but a drawing room, reading and writing room, card room, library, and gymnasium are also provided. The dining saloon, the central portion of which has a maximum height of 31 feet, is two decks lower, the principal entrance being situated centrally. Passenger elevators are provided on each side of the entrance, and, in addition, there are two main staircases—one on each side of the ship. A large swimming bath placed forward of the dining saloon forms another attractive feature of the accommodation. In all three classes, about 4,000 passengers can be carried, 850 of the number being first-class; a number of more or less elaborate suites are provided for the latter in addition to the usual cabins and staterooms. The vessel is propelled by turbines of the Parsons' type direct coupled to four propellers and developing a total of 66,000 shaft horsepower when running at 180 revolutions per minute. The high pressure turbine drives the port inboard shaft, and exhausts into the intermediate pressure turbine driving the starboard inboard shaft. The exhaust from the intermediate pressure turbine is divided equally between the two low-pressure turbines, which drive the two wing shafts; one astern turbine is provided on each of the four shafts. Steam is supplied by 48 water-tube boilers of the Yarrow-Normand type burning oil fuel on the White system and working at about 250 pounds per square inch; the total heating surface of all the boilers is about 220,000 square feet. It is expected that a speed of 25 knots will be reached when the machinery is running normally. The *Majestic*, it will be remembered, was built by Messrs.

Blohm and Voss, of Hamburg for the Hamburg-Amerika Line, and was originally named the *Bismarck*, but before her completion she was handed over to the Reparations Commission and was acquired from them by her present owners. We could wish, of course, that the world's largest vessel had been of British construction, but the fact that she now flies the British flag is a source of considerable satisfaction.—"Engineering," May, 1922.

THYSSEN-HOLZWARTH OIL AND GAS TURBINES.

EXTRACT FROM TREATISE IN GERMAN BY PROFESSOR W. SCHULE, TRANSLATED FOR "MOTORSHIP" BY HANS HOLZWARTH, THE INVENTOR OF THIS UNIQUE INTERNAL-COMBUSTION ENGINE.

The question has to be decided whether or not oil and gas turbines built up to the present time meet the conditions imposed by the operating functions and conditions of modern power work. Scientific considerations, historical comparisons and especially tests made by Professor W. Schüle with the Holzwarth gas-turbine lead him to the conclusion that so far only the Holzwarth gas and oil turbine built by the Maschinenfabrik Thyssen of Mulheim Rühr answers these very difficult conditions—difficult in consideration of the high standing, reliability and efficiency of gas and oil piston-engines. Economically the Holzwarth gas-turbine has already equalled the steam-turbine, and the tests now under way shows a tendency to further increase economy towards that of the gas piston-engine.

Thermodynamically the Holzwarth gas-turbine is superior to the gas piston-engine and the combustion turbine of Armengand Lemale in all degrees of compression. At low compression none of the other processes—it is claimed—can compete with the Holzwarth process. Figure 1 clearly

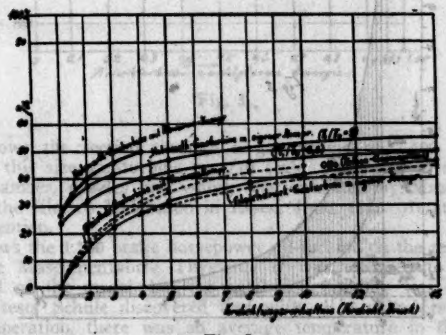


Fig. 1.

indicates this. A thermodynamic efficiency of 45 per cent is attained with the Holzwarth process at a compression ratio of 3, with the combustion turbine at a compression ratio of 10, and with the piston engine at a ratio of 12. In the explosion and combustion turbines air and gas are compressed in a separate auxiliary compression apparatus, while the piston-engine also operates as a compressor.

Therefore the efficiency of compression will be greater with the piston-engine than with the turbines, and it is very important to perform this compression work in turbines at lower stages than in piston-engines, as does the Holzwarth turbine. In all combustion-engines the temperature at the beginning of the expansion process is about 1,500-1,700 centigrade and at the end about 1,000 centigrade. The expansion process is the working process.

Consequently all the working parts have to deal with these temperatures, and the way to withstand these temperatures is utilization of the piston-engine design, the same as in the Holzwarth design. The time during which these high temperatures prevail, must be very short and following it must come a period of low temperature. The rhythmical change between high and low temperatures makes the explosion-turbine of Holzwarth just as safe as the piston-engine. Absence of this rhythmical change makes a combustion-turbine impossible.

Any step to cool these high working-temperatures by mixing steam with the combustion gases is wrong thermodynamically as well as from a practical aspect.

While the steam-turbine requires many stages, according to the drop of steam pressure from 190 pounds per square inch to 28½ inches—a ratio of about 260, the explosion turbine only requires one stage for its drop of pressure from 230 pounds to atmosphere—a ratio of 16. But in the steam-turbines the jet works uniformly, while on explosion turbines it works like a shot with decreasing jet-velocity. Fortunately, this does not matter much as shown by Figs. 2 and 3. In Fig. 2. B. H. C. represents the p. v. diagram

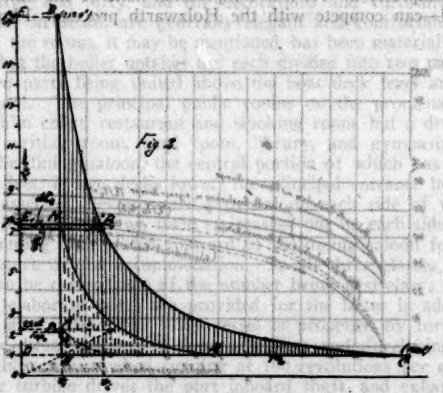


Fig. 2.

of the Holzwarth-turbine working with a compression of 2.2 atmospheres and an explosion pressure of 17.3 atmospheres absolute. While the pressure decreases during the expansion from 17.3 atmospheres absolute to 7:3 atmospheres absolute the part of the energy marked with vertical lines has been transformed into kinetic energy and only the small balance below M.N. is left.

Fig. 3 shows that the jet-velocity decreases slowly until about 90 per cent of the total energy is transformed into kinetic energy. Only during the last 10 per cent, which is not of much importance, the decreasing of the jet-velocity takes place at a quicker ratio. Therefore, the turbine efficiency of the one stage is not much lower than that of a similar steam-turbine wheel. The shocks which the running vanes have to stand in the Holzwarth turbine are, of course, much greater than with steam-turbines. But they easily stand these shocks, due to construction and material.

Fig. 6 shows the method of securing the vanes to the running wheel. The material is soft iron with no nickel or chrome addition.

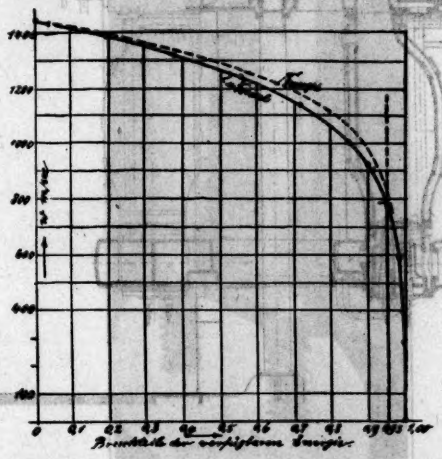


Fig. 3.

Fig. 5 shows the section through a gas-turbine of 1,500 horsepower. A model of this size gas-turbine in 1/10 scale, partly cut to show valves, explosion chamber, nozzles and wheel, was exhibited by Maschinenfabrik Thyssen at the Electro Exhibition in Essen, June, 1920. It attracted considerable attention.

Fig. 7 shows the 1,500 brake horsepower oil-turbine on the test-bed in the plant of the Maschinenfabrik Thyssen. It was intended to exhibit the same instead of the model, but the exhibition building was not suitable. During the tests Schüle discovered that with all 6 chambers of the oil-turbine in operation, there was an average temperature in the explosion chambers of 470-550 Cent. and a temperature of 430-470 Cent. in the turbine wheel housing. Only 9 per cent of the total heat was transmitted into the cooling-water surrounding the explosion chambers. (In Diesel-engines up to 50 per cent is lost in the cooling-water.) This is due to the very short duration of expansion (less than 1/10 seconds) and means that practically all the heat not transformed into energy at the shaft remains in the mixture of exhaust-gas and scavenging-air leaving the oil-turbine at about 450 Cent. It is very essential to make further use of this exhaust heat by producing steam, and driving the little steam-turbine blower.

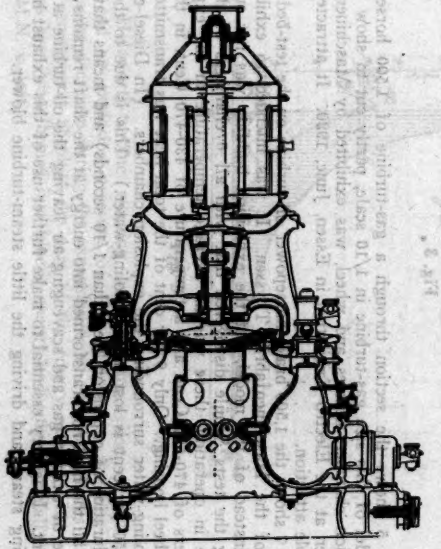


FIG. 4.

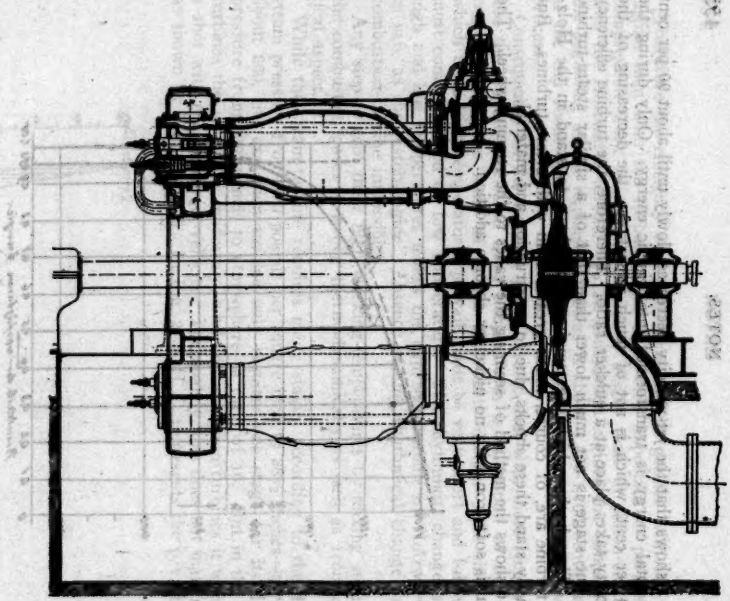


FIG. 5.—SECTIONAL VIEW OF THE THYSSEN-HOLZWARTH 1,500 B. H. P. GAS TURBINE.

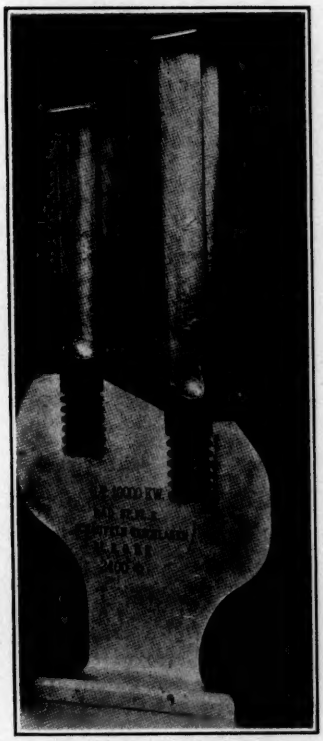


FIG. 6.—METHOD OF SECURING THE VANES TO THE RUNNING-WHEEL.

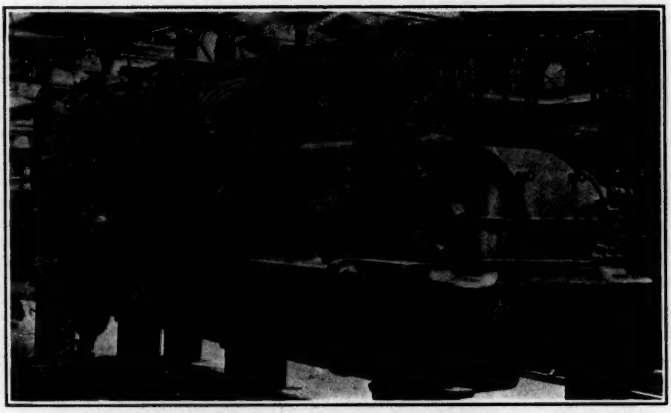


FIG. 8.—THE 500 B.H.P. THYSSEN-HOLZWARTH OIL TURBINE, WHICH MAY BE THE POWER OF THE FUTURE FOR MERCHANT SHIPS.

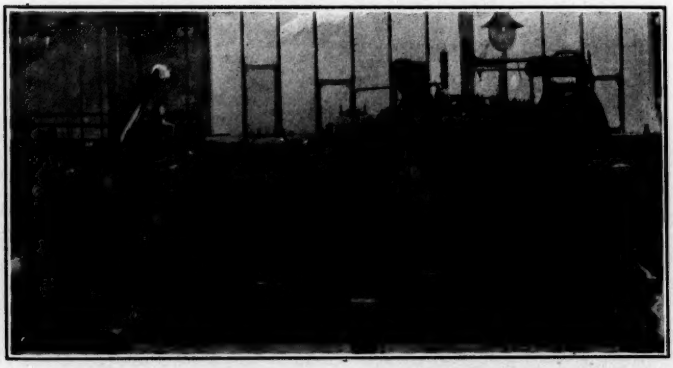


FIG. 9.—THE 1,000 HORSEPOWER THYSSEN-HOLZWARTH GAS TURBINE.

With the vertical 1,000-horsepower Thyssen experimental gas-turbine Schüle has already made several tests, and is still carrying on experiments. He reports that he found the machine in an absolutely safe working condition, and that he could carry through his tests without any trouble. With a compression of 2.2 atmospheres absolute he found with a wheel having but one set of vanes an efficiency of 25 per cent when developing 1,200

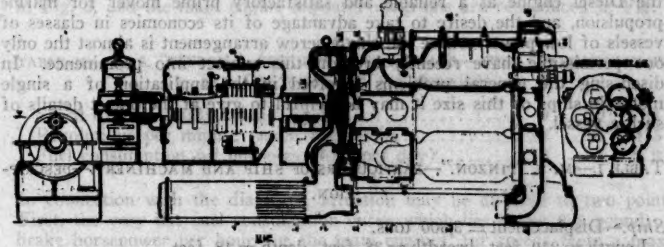


Fig. 7.—Section of the Thyssen-Holzwarth 1500 b.h.p. oil turbine

horsepower. Further tests show a tendency of an increasing effective efficiency and Schüle concludes with the remark that the time will soon come when the effective efficiency of the Holzwarth gas-turbine will have reached a position halfway between the steam-turbine and the gas piston-engine. In developing the Holzwarth turbine to a practical engine of the highest economy the Maschinenfabrik Thyssen will produce a new and powerful unit for world economy of energy.—"Motorship," May, 1922.

LARGEST MARINE DIESEL ENGINE.

Under construction at the works of Sulzer-Freres, Winterthur, Switzerland, is a marine-type Diesel engine of 3,500 shaft horsepower in six cylinders. This engine when completed will hold for a short time the record for power, although several larger engines are being completed elsewhere, in addition to the three naval engines of 4,000 shaft horsepower at the same plant. Sulzers have also built two 4,000 brake horsepower stationary Diesel engines.

In the 3,500 brake horsepower Sulzer engine the crankshaft has a total length of 53 feet ¼ inch, and is in four parts coupled together. Each of the two middle sections has three cranks of 23.3 inches radius, set at an angle of 60 degrees to each other.

Sulzer Brothers have adopted the system of scavenging by air from turbo blowers, driven independently of the main engines by electric motors. While at sea, this means that an auxiliary engine must be run to supply electric current, but it frees the main engines entirely from the work of driving the scavenging pumps, thus increasing the horsepower actually delivered to the shaft. The main crankshaft is also shorter than it would otherwise be, as no cranks are necessary for the scavenging pumps, and the space required for the engines is thereby much reduced.

Both these points are of great advantage in the case of the larger marine engines which will be required in the near future as the motorship becomes more and more generally adopted.—"The Steamship," May, 1922.

DIESEL MACHINERY FOR SINGLE-SCREW MOTOR SHIPS.*

By JAMES RICHARDSON.

Notwithstanding the vast increase in the number of ships fitted with internal combustion engines, single-screw motor ships of 2,000 tons deadweight and over have hitherto been exceptional. Increasing confidence in the Diesel engine as a reliable and satisfactory prime mover for marine propulsion, and the desire to take advantage of its economies in classes of vessels of low power, where the single-screw arrangement is almost the only one permissible, have recently brought this subject into prominence. In discussing the general problems involved in the application of a single motor in ships of this size it may be helpful to give at the outset details of such a vessel.

TABLE I.—M. S. "PINZON."—PARTICULARS OF SHIP AND MACHINERY.—DESCRIPTION.

Ship.—Displacement = 3,300 tons.
Length = 240 feet, breadth = 38 feet, depth = 18 feet.
Winches.—Six fitted, each 16 horsepower.
Windlass.—One fitted, 50 horsepower.
Main Engine.—Beardmore-Tosi four-cycle, single-acting, six-cylinder, 620 mm. (24⅜ in.) bore, 975 mm. (38⅜ in.) stroke, 1,250 brake horsepower at 120 revolutions per minute.
Main Injection Air Compressors.—Two three-stage. Swept volume per revolution one compressor = 2 cubic feet.
Swept volume per revolution two compressors = 4 cubic feet.
Auxiliary Compressor.—Three-stage. Capacity first speed = 88 cubic feet per minute free air.
Capacity full speed = 176 cubic feet per minute free air.
Emergency Compressor.—Swept volume = 12 cubic feet per minute free air.
Starting Air Receivers.—Three at 70 cubic feet each = 210 cubic feet.
Maximum working pressure 500 pounds per square inch.
Main Engine Blast Bottle.—One working, one spare fitted.
Capacity of one bottle = 5 cubic feet.
Maximum working pressure 1,000 pounds per square inch.
Auxiliary Generators.—Two fitted, each 50 kilowatts, Diesel driven.
Engines of trunk piston type, two-cylinder, four-cycle, 75 brake horsepower, 250 revolutions per minute.
Compressors (Auxiliary Generators).—For one generator engine, swept volume of compressor at 250 revolutions per minute = 41.2 cubic feet per minute free air.
Two generator engine, swept volume of compressor at 250 revolutions per minute = 82.4 cubic feet per minute free air.
Starting Bottle (Auxiliary Generators).—Two per engine, capacity = 3.54 cubic feet each.
Maximum pressure 1,000 pounds per square inch.
Blast Bottle (Auxiliary Generators).—One per engine, capacity = 1 cubic foot each.
Maximum pressure 1,000 pounds per square inch.
Propeller.—Diameter = 12 feet 6 inches, pitch = 11 feet.

* Institution of Naval Architects. Abstract.

Table I gives the leading characteristics of the machinery and equipment of the single-screw motor ship *Pinzon*, built and engined by William Beardmore and Co., Limited, of Dalmuir.* All deck and engine-room auxiliaries are electrically driven. No boiler is fitted. The total consumption of fuel oil for all purposes in this ship at 9, 10, and 11 knots at sea, fully laden, in reasonably good weather, is 3, 4, and 5.2 tons per day, respectively, as given in Table II. The results of the machinery trials are shown by the curves in Fig. 2, and the main engine indicator diagrams are reproduced in Fig. 1.

TABLE II.—M. S. "PINZON."—RESULTS OF SEA TRIALS.—FUEL CONSUMPTION PER DAY (IN TONS) FOR ALL PURPOSES.

Speed of ship (fully laden), knots	9	10	11
Revolutions per minute (main engines)	95	105	115
Fuel consumption, all purposes (tons per day)	3	4	5.2

In connection with the diagrams, attention may be directed to two points. First, the very flat fuel consumption curve which lies below 0.5 pound per brake horsepower per hour for the main engines, for speeds from a maximum of 125 revolutions per minute to 81 revolutions per minute, *i. e.*, from over 1,400 brake horsepower to 420 brake horsepower. Secondly, an inspection of all the indicator cards shows from the toe of the diagram the very rapid release of the exhaust gases. The pressure drops to atmospheric almost before the bottom dead center is passed. This result is due to the large valve areas possible with the well-known Tosi arrangement of combined inlet and exhaust valves with the director valve controlling the exhaust gases and induction air. A further feature of this system is the extremely low exhaust temperatures, which never exceed 560 degrees F., although measured in the stream of the gases in the cylinder head.

Broadly stated, above 7,000 tons deadweight and 11 knots speed there is a good case for the retention of the standard twin-screw arrangement of oil engines because of the lower total cost, the reduced weight of machinery, the shorter length of engine-room, which compensates for the cargo space taken up by the extra shaft tunnel, and the reduced risks of breakdown. The *personnel*, however, is increased.

Turning in detail to the problems inherent to single-screw Diesel machinery, it may be said first that the dictates of reliability and the limited man power available on board demand that certain operations shall be made possible and certain conditions fulfilled.

(1) The crank shaft shall be capable of being removed and replaced with the minimum dismantling of the main engine parts. As an alternative, the after length of the crank shaft, being the more liable to breakdown, shall be so accessible that it can be readily removed and replaced.

(2) That part of the fuel injection air service common to all the main engine cylinders should be in duplicate, including the blast bottle, so that failure of any pipe, joint, valve or chest, &c., shall not necessitate, at most, more than a momentary stoppage.

(3) Arrangements must be made so that all the valves in the cylinder head can be rotated on their seats by hand.

(4) All the main parts, including pistons, must be completely accessible.

(5) The failure of any one or two fuel pumps, supplying fuel under pressure to the main cylinders, must not necessitate a stoppage.

* See "The Engineer," February 10th, 1922.

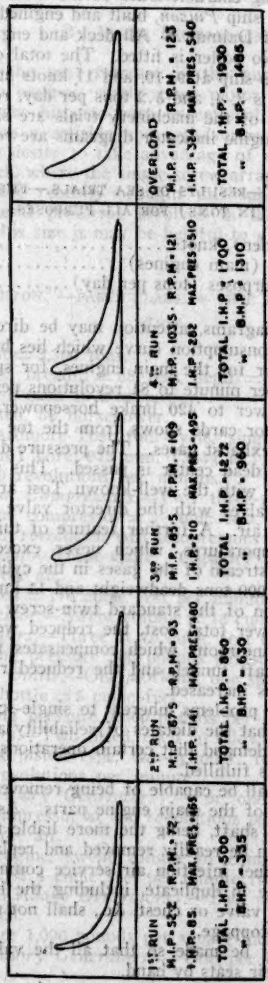

1ST RUN
M.I.P. = 5252 R.P.M. = 72
I.H.P. = 3515 MAX. PRES = 485
TOTAL I.H.P. = 500
" B.H.P. = 335
2ND RUN
M.I.P. = 87·5 R.P.M. = 93
I.H.P. = 1461 MAX. PRES = 480
TOTAL I.H.P. = 860
" B.H.P. = 630
3RD RUN
M.I.P. = 85·5 R.P.M. = 109
I.H.P. = 210 MAX. PRES = 495
TOTAL I.H.P. = 1272
" B.H.P. = 960
4TH RUN
M.I.P. = 103·5 R.P.M. = 121
I.H.P. = 282 MAX. PRES = 510
TOTAL I.H.P. = 1700
" B.H.P. = 1310
OVERLOAD
M.I.P. = 117 R.P.M. = 123
I.H.P. = 324 MAX. PRES = 540
TOTAL I.H.P. = 1930
" B.H.P. = 1485
The Engineer
Swan Sc.

FIG. 1.—INDICATOR CARDS TAKEN DURING SPEED TRIALS OF THE MOTOR SHIP PINSON

(6) The subject of main engine flexibility, the maneuvering gear, valve setting, and the proportioning of the flywheel, and the whole question of starting or maneuvering air supply must have greater attention than in the case of a twin-engined vessel.

(7) The engine shall be capable of being readily maneuvered by one operator.

As with steam installations, so with motor machinery; the subject of the auxiliaries, both deck and engine-room, is only now receiving the attention demanded by its vital importance. With motor ships the rival claims of an electric or steam drive still form, and are likely to continue to prove, debatable ground. The alternative of an electric engine-room and steam-driven deck machinery is possible and may commend itself in certain cases. In Table III is given a comparison of the consumption of fuel in similar vessels, one a steamer and the other a motor ship, on the same trade. The total consumption of coal is five times that of oil (by weight) in the latter taken over a whole voyage, and is achieved by the utilization of all electric machinery for deck and engine-room auxiliaries, deriving current from Diesel-driven auxiliaries. With steam auxiliaries on the motor ship, on this trade, the fuel consumption would only be one-third (not one-fifth) that of the steamship.

TABLE III.—M. S. "PINZON" AND SIMILAR STEAMSHIP.—COMPARISON OF FUEL CONSUMPTIONS AT SEA AND IN PORT.

Steamship.
Triple-expansion engines, single screw.
Ship, 270 feet by 38 feet by 18 feet; draught, 17 feet 5 inches.
Voyage, London to Spanish port.
Indicated horsepower, 1,073.
Revolutions per minute, 75.8.
Speed of vessel, 9.35 knots.
Coal used per day for all purposes, 17.42 tons.
Coal used by auxiliaries per day, 2.5 tons.
Coal used in port per day, discharging and loading, 3 tons.
Banked fires, used per day, sixteen days on average voyage, 1 ton.

Motor Ship Pinzon.
Diesel engines, single screw.
Ship, 240 feet by 38 feet by 18 feet; draught, 17 feet 6 inches.
Voyage, Glasgow to Spanish ports.
Diesel indicated horsepower, 1,290.
Equivalent steam indicated horsepower, 1,170.
Revolutions per minute, 106.
Speed of vessel, 10.2 knots.
Oil used per day for all purposes, 4.5 tons.
Oil used by auxiliaries per day, 0.2 ton.
Oil used in port per day, discharging and loading, 0.37 ton.

Table III emphasizes clearly the importance in a steamer of the fuel consumption attributable to the auxiliaries generally, and in such conditions as in a coasting trade, in particular, the fuel consumption of the deck machinery. Whereas at sea the ratio of coal to oil for all purposes is 4 to 1, in port this ratio is over 10 to 1 if fires are "banked." A small oil-fired boiler is generally required for the heating of the accommodation and of the oil fuel, although the former can be carried out by electric radiators, and if only a small quantity of oil is carried exhaust gases can be utilized for heating it. There is probability of the general adoption in future of a

reasonable utilization of waste heat with Diesel marine installations, whereby oil fuel and accommodation heating can be carried out entirely by the main engine exhausts when at sea and partially, at any rate, by the auxiliary exhausts when in port. The subdivision of the requisite total Diesel electric generator capacity and the desirable margin to be legislated for is an interesting point. Sometimes two generator engines and in a number of twin-screw ships four have been fitted. The actual electric loads on the M. S. *Pinzon* under various conditions have been tabulated and the load in kilowatts on the auxiliary Diesel generators may be summarized as follows:—Normal sea load: maximum, 59.85; minimum, 23.2; maneuvering load: maximum, 151.25; minimum, 46.85; port load: maximum, 72.2; minimum, 22.6. In general, three sets are preferred, one to be in action normally at sea and in port and two when maneuvering or working cargo fully, one serving as a stand-by. Three such sets of 50 kilowatts each meet the case shown in the table, and only rarely does the minimum load fall substantially below half the power of one unit, a proportion desirable for good working conditions with such machines.

The maneuvering load is the highest, even when it may be arranged that the auxiliary compressor is slowed down if and when the windlass load comes on, and that the radiators may be cut out if necessary. The maneuvering load comes within the capacity of two generators. The greatest individual load is the auxiliary compressor required to make up starting air. This machine is of sufficient capacity to supply the main engine with injection air at full power in emergency, should both the main engine-driven injection compressors be out of action.

The conservation and replenishment of starting air is a subject of vital importance with internal combustion machinery, as the ability to maneuver the ship is lost if a sufficient quantity and pressure of air is not always available to start the main engine. Apart from the subject of conservation, reasonably rapid replenishment must be arranged for, as it is possible for a large quantity of air to be lost, as, for instance:—

(*i*) If the exhaust valve of the cylinder into which starting air is being admitted should be hung open;

(*ii*) If one or more starting air valves in the cylinder head hang open;

(*iii*) Or if the engine maneuvering gear should jamb in the six or three cylinders on air position and other means not be immediately adopted to shut off the air.

As an example of the demand made upon the starting air storage capacity, two extracts from the log of the M. S. *Pinzon* is given (Table IV) when entering Dalmuir Basin. There is one characteristic of these extracts and similar entries in the log to which reference might be made, and that is the prevalence of the repetition of the *motif*, "slow ahead, stop, slow ahead, stop," having the effect of propelling the ship by repeated thrusts rather than by a sustained effort. It might be suggested that the minimum sustained power possible with the main Diesel engine is excessive. This is not so. The engine has been run for considerable periods at 24 revolutions per minute corresponding to a maximum speed through the water of 2.5 knots, when full way has been obtained, and a piston speed of 150 feet per minute, giving a ratio of maximum to minimum piston speed, or speed of revolution of 122.5/24 or 5.2 to 1. It is difficult to foresee any substantial improvement upon this result, since regular firing of the very minute charges of oil fuel can hardly be expected at a lower piston speed without some substantial increase in complication, as would be involved, for instance, by preheating the induction air. This extremely and exceptionally low speed of

TABLE IV.—M. S. "PINZON."—COPY OF LOG OF ENGINE TELEGRAPH ORDERS CARRIED OUT ON JANUARY 13TH, 1922, ENTERING DALMUIR BASIN.

	p. m.		p. m.
Half ahead	4.57	Stop	6.05
Slow ahead	4.59	Slow ahead	6.1
Half ahead	5.4	Stop	6.2
Slow ahead	5.12	Slow ahead	6.4
Stop	5.14	Stop	6.4.5
Slow ahead	5.15	Slow ahead	6.5
Half ahead	5.26	Stop	6.5.1
Slow ahead	5.26.5	Slow astern	6.5.5
Half ahead	5.37	Full astern	6.6
Slow ahead	5.40	Slow astern	6.6.5
Stop	5.46	Stop	6.7
Slow ahead	5.47	Slow ahead	6.7.1
Stop	5.48	Slow astern	6.7.2
Full astern	5.48.5	Full astern	6.7.5
Slow astern	5.50	Slow astern	6.7.5
Ahead slow	5.51	Stop	6.7.6
Full astern	5.51.5	Slow ahead	6.7.7
Slow astern	5.52	Stop	6.8
Stop	5.52.5	Slow astern	6.8.5
Full astern	5.53	Stop	6.8.6
Slow astern	5.53.1	Half astern	6.9
Stop	5.53.5	Slow astern	6.9.1
Astern slow	5.54	Stop	6.9.2
Stop	5.54.5	Slow ahead	6.9.5
Astern slow	5.55	Stop	6.10
Stop	5.55.5	Slow ahead	6.12
Slow ahead	5.56	Stop	6.12.5
Stop	5.59	Finished with engines	6.15
Slow astern	6.0		

piston of 150 feet per minute is attained by correct combustion-producing means and by the fitting of a fly-wheel with suitable momentum effect.

As is now recognized by Lloyd's Register, the fitting of a fully proportioned fly-wheel isolates the thrust, intermediate and propeller shafts from the variation of engine turning moment. The crank shaft is also equally protected from shocks, which might otherwise be transmitted from the propeller. Such a fly-wheel has no deleterious effects when starting up or reversing. Momentum is first given by an elastic medium—compressed air—and during a sudden reversal which can be carried out in less than ten seconds, with full way on the ship, the starting air on being admitted astern, if the engine compression has not already brought the engine to a stop, gently pulls the engine up and gives the initial impulses in the opposite direction.

The two diagrams reproduced in the right hand top corner of Fig. 2 show clearly the only remaining justification with modern Diesel engines for the charge of lack of flexibility. When starting up, especially with a view to doing so on the minimum expenditure of compressed air, fuel injection is commenced early, and the engine accelerates to "half speed" before retarding to "slow." The total amount of impulse given to the ship is, therefore, approximately twice the ideal. The reconciliation, therefore, of the two

somewhat conflicting aims of minimum compressed air used and minimum speed of revolution during starting, without undue complication of the maneuvering gear, is worthy of close study.

It will be found with every type of engine that there is a pressure of compressed air most suitable for starting with the minimum expenditure of air storage energy.

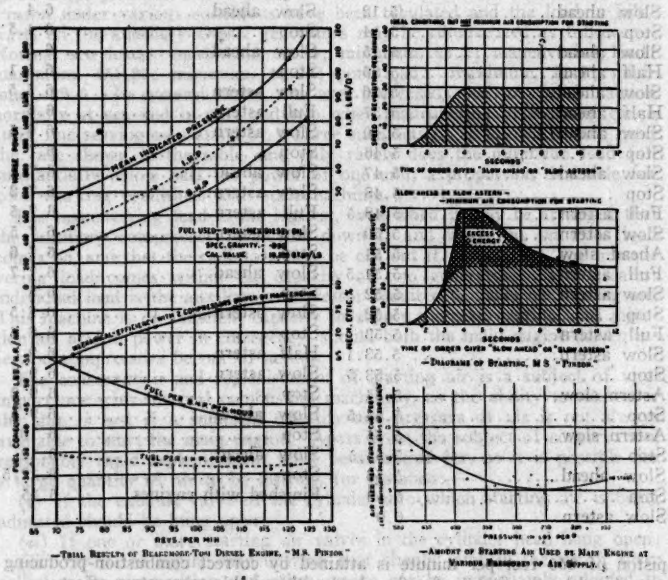

MEAN INDICATED PRESSURE
L.H.P.
B.H.P.
FUEL USED—SHELLMEX DIESEL OIL
SPEC. GRAVITY. .894
CAL. VALUE 19,500 B.T.U./LB
MECHANICAL EFFICIENCY WITH 2 COMPRESSORS DRIVEN BY MAIN ENGINE
FUEL PER B.H.P. PER HOUR
FUEL PER I.H.P. PER HOUR
REVS. PER MIN
SECONDS
TIME OF ORDER GIVEN "SLOW AHEAD" OR "SLOW ASTERN"
EXCESS OF AIR USED
—Trial Results of Blackwood-Tom Diesel Engine, "M.S. Pinzon"
—Diagrams of Starting, M.S. "Pinzon."
—Amount of Starting Air Used by Main Engine at Various Pressures of Air Supply.

FIG. 2.—CURVES SHOWING PERFORMANCE OF PROPELLING MACHINERY OF THE "PINZON."

The bottom right hand curve given in Fig. 2 shows that with pressure between 250 pounds and 350 pounds per square inch the loss of air is a minimum. Whatever the storage pressure—500 pounds per square inch with the *Pinzon*—a reducing valve to ensure that the pressure on the engine side of the air supply is not greater than 350 pounds per square inch, or other means such as working on one of a number of reservoirs, will facilitate the conservation of air.

To consider the replenishment of the storage, Appendix I gives data obtained. Making the assumption that one start per minute will be the maximum required, over any appreciable length of time, and as the receivers fitted have a capacity of forty starts without re-charging—which will cover any exceptional number of starts called for during a short period—it is shown, and has been proved in practice, that:—(1) Neglecting the excess injection air from the main engine, amounting to 2 cubic feet of free air per revolution of the main engine, there is available air storage and

means for replenishment capable of sustaining one maneuver per minute continuously. (2) If the main engine makes 75 revolutions each move, its excess injection air on being tapped off to the starting bottles will restore what has been taken for the start.

Analyzing a number of logs of "stand-bys," it is found that the average of one maneuver per minute is not exceeded. The average "move," however, is of much less than seventy-five revolutions, so that actual conditions are a combination of (1) and (2) above, but condition (1) must be satisfied in order:—

(*a*) To give a stand-by in case of breakdown, with the main engine injection compressors.

(*b*) To cope with the demands for air for starting during a short spell of maneuvering, calling for more than one move per minute, if air has been lost for any reason.

(*c*) To supply starting air when the main engine injection compressor is either out of action or is working inefficiently and giving no excess air to the starting receivers.

The starting and reversing of an engine operating on the four-cycle principle have been brought to the simplest terms, consistent with operation from the usual starting platform. Whatever may be permissible with a twin-screw ship, it is considered essential with a single-screw installation that the controls should be on the engine-room floor level, and the major portion of the gear serves to bring the controls to the usual location. Single-screw Diesel machinery for the average cargo carrier will be much developed in the near future, and the lines of design which will be followed will tend decreasingly to demand a compromise between the best propeller and a smaller and more compact engine, especially with the larger ships. Speeds of revolution will become standardized as with the steam engine, and longer stroke engines will be built. The weight of machinery, the space occupied and the cost will be greater, which will be more than compensated for by the extra accessibility, a built-up crank shaft and a considerably better average propeller performance.

Appendix.

M. S. Pinzon.—Data Relating to Compressed Air—Time to Start from Conditionally "No Air in Ship."

(*a*) When running the emergency compressor of 12 cubic feet per minute swept volume.

Time to fill one blast bottle and one starting bottle for auxiliary generator engine to

800 pounds per square inch = 26 minutes 50 seconds.

1,000 pounds per square inch = 35 minutes 30 seconds.

(*b*) When running the auxiliary compressor.

Time to fill one starting air reservoir to 500 pounds per square inch = 19.5 minutes.

Additional time to top up one blast bottle from 500 pounds to 1,000 pounds per square inch = 2.5 minutes.

(*c*) Air available when maneuvering main engine.

Air from one main engine-driven compressor—the other being sufficient for blast supply = 2 cubic feet per revolution.

Excess air from auxiliary engines (estimated at one-third of air made) = 27.5 cubic feet per minute.

From auxiliary compressor = 176 cubic feet.

(d) Assuming no air is taken from main compressors, we have 174 cubic feet free air per minute available. This is equivalent to approximately 7 cubic feet air at 350 pounds per square inch per minute. The main engine uses from 5 to 7 cubic feet of air at 350 pounds per square inch per start or reversal, so that there is sufficient air available to allow of one maneuver per minute continuously, neglecting any surplus air made by main engine.

(e) 70–75 revolutions of main engine supply air for each maneuver (one compressor used wholly for blast), so that if the main engine is allowed to make 75 revolutions for every maneuver the starting air can be maintained by the main engine without using any auxiliary supply.—"The Engineer," April, 1922.

SURPLUS ELECTRICAL ENERGY FOR STEAM PRODUCTION.

Continental engineers have given considerable attention to the problem of the electric steam boiler, both as a steam producer for industrial process work and as a means whereby surplus electrical energy may be economically used. In our issue of October 14th we described in detail a boiler of French design, used in connection with the paper-making industry. We give in what follows some particulars of another large boiler which was recently described in the German Technical Press.

Made by the Wärme-Akkumulatoren-Gesellschaft, of Berlin, this boiler was specially designed to utilize surplus current during the night and at other light-load periods, the steam generated being supplied to a battery of cellulose digesters at a pressure of from 49–50 pounds per square inch. The boiler illustrated in Fig. 1 is of 6,000 kilowatts capacity, and was designed to work with three-phase current at a pressure of 6,000 volts. Such a boiler may be attached to an already existing battery of coal or oil-fired boilers, its purpose being to utilize automatically any surplus energy that may be available in the network from which the electrical supply is taken. Thus, if supply is drawn from a power station with a combined power and lighting load, the boiler will take up automatically the difference in load between the sum of the lighting and power loads and the full-load capacity of the plant. This is made possible by altering the level of the water in the boiler within very small limits to suit any change in load. With this method of regulation all moving parts inside the boiler are avoided and the complete control arrangements, as shown, are mounted on the outside.

With the type of electrode used in the French boiler, similar to that shown in Fig. 2, regulation is only possible to suit the load on the boiler, and does not take into account the surplus energy available. The disadvantages of this type of electrode are that, particularly with small boilers, there being no feed control, phase disturbances are likely to be caused, which may cause flickering in the lamps on the adjoining lighting circuits. Eddies also are frequently set up round the electrodes, which in such case are often alternately laid bare and then quickly covered with water, causing unequal loading.

Quartz insulation has been tried for these electrodes, but it has been found that in spite of its very low expansion coefficient and its insensibility to temperature changes, it does not answer well. It is much more brittle than porcelain, and is quickly corroded by boiler water. In the boiler described the type of electrode used is not influenced by the specific conductivity of the water within wide limits, and all kinds of water, including chemically purified water, may be used.

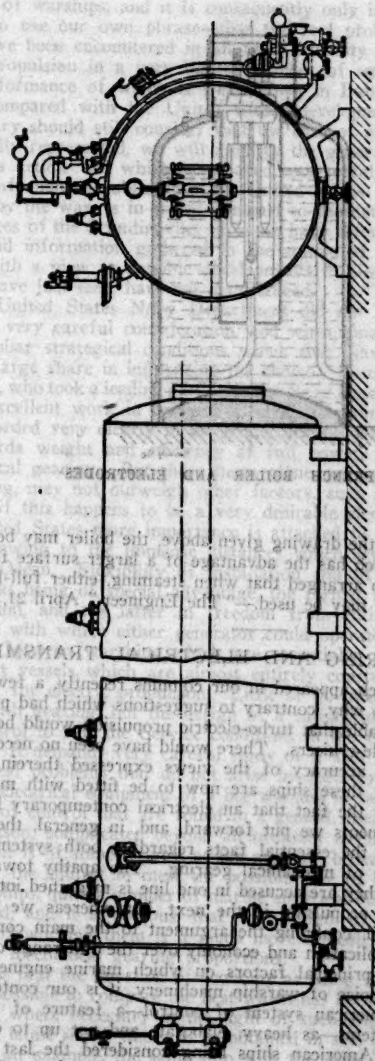


FIG. 1.—BOILER OPERATED WITH THREE-PHASE CURRENT

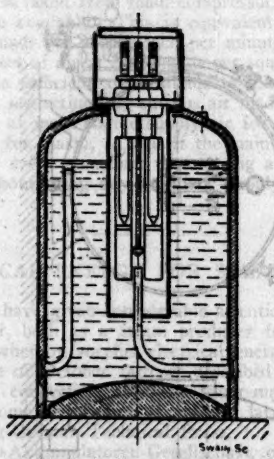


FIG. 2—FRENCH BOILER AND ELECTRODES

As will be seen from the drawing given above, the boiler may be built in the horizontal form, which has the advantage of a larger surface for steam raising. The boiler is so arranged that when steaming, either full-load current or overload current may be used.—"The Engineer," April 21, 1922.

MECHANICAL GEARING AND ELECTRICAL TRANSMISSION.

In a brief article which appeared in our columns recently, a few leading reasons were given as to why, contrary to suggestions which had previously appeared, it was improbable that turbo-electric propulsion would be adopted for the new British battle-cruisers. There would have been no necessity for calling attention to the accuracy of the views expressed therein, as evidenced by the fact that these ships are now to be fitted with mechanical gearing, were it not for the fact that an electrical contemporary has since attacked both the statements we put forward, and, in general, the attitude of those who, with all the essential facts regarding both systems before them, adhere to the use of mechanical gearing. The apathy towards new developments of which they are accused in one line is magnified into a campaign against electrical propulsion in the next, but whereas we refrained from criticisms of detail, confining the argument to the main comparative features of weight, complication and economy over the full range of power, which are perhaps the principal factors on which marine engineers must necessarily base their choice of warship machinery, it is our contemporary which describes the American system of control—a feature of vital importance in electric systems—as heavy, elaborate and not up to date, and which objects "to the American ships being considered the last thing in electric propulsion." They are, from the *New Mexico* to the *Lexington* class, as we may remind the writer, the only examples of the electrical pro-

pulsion of warships, and it is consequently only in "these remarkable vessels"—to use our own phrase—that the real problems of electric propulsion have been encountered in all their diversity and magnitude, for electrical propulsion in a very limited number of small Scandinavian vessels, the performance of which is well known in Baltic circles, is on a trivial scale compared with the United States developments. But lest our contemporary should still consider that the case of electric propulsion has not been fully represented, we will amplify the general statement made in our previous article and which was based, not merely on the extraordinarily ample information published in the United States regarding these vessels—which, by the way, is in strong contrast to the details available of the performances of the Scandinavian ships we have referred to—but also on very first-hand information gathered in the course of an investigation recently made with a view to possible developments in electric transmission, which, as we have just seen, have not materialized.

The United States Navy Department did not adopt electric propulsion without very careful consideration, and when, finally, it was decided upon, the peculiar strategical conditions which that Navy has to meet naturally bore a large share in influencing the choice. Commander S. M. Robinson, U. S. N., who took a leading part from the Navy Department side and has done much excellent work in bringing electric transmission to its present state, has recorded very clearly in an article dealing with the subject, that both as regards weight and efficiency at full power, the advantage lies with mechanical gearing. But these items alone, unless the advantage be overwhelming, may not outweigh other factors, such as economy at very low speeds, if this happens to be a very desirable element, and in the case of the United States more importance is attached to it than in Great Britain, largely owing to the immense distances which separate their bases in the Pacific Ocean. Flexibility of installation and of operation are also considered as being of special advantage, the former in admitting of elaborate subdivision, and the latter in freedom from entire breakdown, owing to the ease with which either generator could operate any shaft motor. The case as between warships which are essentially variable-speed ships, and merchant vessels which are almost entirely constant-speed ships, is vastly different for electric propulsion. Little or no modification in design is entailed with gearing, but the design of a satisfactory form of transmission and control to suit the Service conditions in the former is a very complex matter. It demands very careful study of a wide number of electrical and mechanical problems that do not arise in ordinary generating work at all, and the electrical equipment of a big warship to satisfy the same conditions of ease of control that are found in a geared installation must be, for that very reason, complicated beyond all ordinary electrical work. This particular point is of some interest at the moment, because, apart from the study made of the subject to meet the possible case of an alternative design for the big ships, quite a number of marine electrical inquiries have recently been circulated, and it is obvious that the design of generators, motors and system of working them is a much more difficult proposition than that of selecting mechanical gearing proportions.

Taking first the question of economy as being of primary importance, it is only possible to compare the best which has been accomplished up to date, modified where necessary in the light of such improvements as experience with the latest shows likely to be obtained in sets under construction. Taking, for an example, the same ship with the same boilers and shafting, &c. we have the two stages of heat conversion into power with each form of transmission; one in the turbine, and thence by gearing or electric transmis-

sion. Single reduction mechanical gearing of the type generally adopted in Great Britain has, over a wide range of power, an efficiency up to 98.5 per cent, and even down to very low powers the loss does not exceed 3 to 4 per cent. The efficiency of the generators in the *New Mexico* and *Maryland* is given officially and by the builders as ranging from 95 per cent at 10 knots and 96 per cent at 15 knots, to 97 per cent at full power, while the motor efficiency is fairly constant at 95.5. The comparative transmission loss between turbine and shaft is—

Percentage of full power	One-tenth	One-half	Full
Generator efficiency	95.0	96.0	97.0
Motor efficiency	95.5	95.5	95.5
Combined efficiency	90.7	91.7	92.6
Efficiency of mechanical gearing	96.0	97.0	98.0
Per cent superiority of gearing, per cent	5.3	5.3	5.4

These figures were in use for responsible comparisons of the two systems within the last month or two.

Next, what type of turbine can or is to be adopted? With single reduction gearing, two cylinders in series with two pinions working on a large common gear wheel offer the best results for economy and space occupied, and with two cylinders a very high turbine "K" is being easily and cheaply attained; in fact, one giving a thermal efficiency ratio of 70 per cent,* or a consumption of 9.6 pounds per shaft horsepower, with steam at 250 pounds gauge pressure and 28.5 inches vacuum, which are the guarantee conditions—for dry steam—in the United States ships, is common. With a single cylinder impulse turbine, as in the United States ships under discussion, this high economy is not so easily obtained without considerable increase of size of turbine. The *New Mexico's* turbo-alternators were some 35 feet long overall—they ran at nearly 2,000 revolutions per minute—by 10 feet wide by 12 feet 6 inches high. Their thermal efficiency ratio at full power related to the propeller shafts was about 57.8 per cent.* To improve this materially will involve either a considerable lengthening of machine and increase of weight, or else the higher consumption due to the single cylinder impulse design and its low transmission efficiency must be accepted. For much larger units, particularly if the power or the revolutions are increased, we shall expect to see two cylinders, the low-pressure of which must be double flow, as in the *Hood* and the British destroyers, with consequent considerable increase in weight and overall length. It is quite impossible to dissociate the turbine design from the electric drive. The steam consumption of the *New Mexico*—with the above transmission loss, of course, included—was as follows:—

Speed, knots.	S. H. P.	Lb. per S. H. P. Guaranteed.	Actual.
10	3,200	15.38	14.35
15	9,200	11.57	11.6
19	20,750	11.32	11.5
21.5	33,000	11.53	11.58

The desired consumptions can be varied according to the design of turbine, so as to give good low power economy with the lowest consumption per shaft horsepower at about 80 per cent of full power, that at full power rising slightly above this, or the maximum economy can be arranged for

* In turbine brake horsepower—*i. e.*, without gearing—these values are 71.5 and 62.4 respectively. Both are often exceeded.

full power, in which case the lower power rates will rise somewhat. The former design is adopted for warships, the latter for merchant vessels and the variation in the former is a matter of requirement. In the American ships, with the object of securing as good economy as possible at low powers, the full-power rate was sacrificed. The total consumption per hour of the *New Mexico* is shown in Fig. 1. With the two-cylinder arrangement

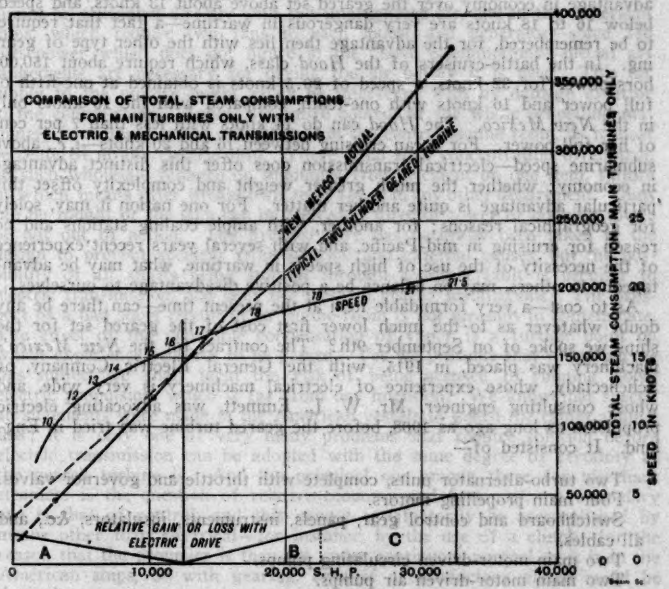


FIG. 1.—STEAM CONSUMPTION FOR ELECTRICAL AND MECHANICAL TRANSMISSION IN SHIPS.

and mechanical gearing, and a thermal efficiency ratio of 67 per cent under the same conditions of pressure and vacuum a consumption at full power of 10 pounds per shaft horsepower is now obtained. The total consumption of such a set is also shown, the figures following the straight Willans line as usual. That for the *New Mexico* falls direct into the origin, but in practice both curves tend to converge slightly below one-tenth full power. The two lines cross at 16.25 knots, or at about one-third of full power, when all four shaft motors are driven from one generator, the other turbo-alternator being cut out, and, like the astern turbines in the geared sets, being kept ready for instant operation, a matter of small moment. The relative gain or loss in steam consumption for the respective main engines is shown in the lines at the foot of the diagram. It is to portion A that so much importance is attached in the United States; portion B is equal to A, and if the vessel spent equal hours at all the different speeds, portion C would represent clear

gain for the geared set. The saving at the upper end of the curve goes a long way in nullifying the advantage of 10 knots. Much depends on what importance is attached to the gain at low powers. In the constant speed ship, the superior transmission efficiency of the gearing must inevitably restrict the adoption of any less efficient form of drive, but it is, *par excellence*, in the battle-cruiser type that the advantage of electric transmission is seen. As shown in the *New Mexico's* speed curve, there is little tangible advantage in economy over the geared set above about 13 knots, and speeds below 16 to 18 knots are very dangerous in wartime—a fact that requires to be remembered, for the advantage then lies with the other type of gearing. In the battle-cruisers of the *Hood* class, which require about 150,000 horsepower for 32 knots, a speed of 20.5 knots is obtained at one-fifth of full power and 16 knots with one-tenth, compared with the 10 knots only in the *New Mexico*. The *Hood* can do 14 knots with less than 7 per cent of her full power. For ocean cruising between 16 and 20 knots—*i. e.*, above submarine speed—electrical transmission does offer this distinct advantage in economy; whether the much greater weight and complexity offset this particular advantage is quite another matter. For one nation it may, solely for geographical reasons; for another, with ample coaling stations and no reason for cruising in mid-Pacific, and with several years recent experience of the necessity of the use of high speeds in wartime, what may be advantageous to others, may on balance be a positive disadvantage to ourselves.

As to cost—a very formidable item at the present time—can there be any doubt whatever as to the much lower first cost of the geared set for the ships we spoke of on September 9th? The contract for the *New Mexico's* machinery was placed, in 1915, with the General Electric Company, of Schenectady, whose experience of electrical machinery is very wide, and whose consulting engineer, Mr. W. L. Emmett, was advocating electric propulsion as long ago as 1908, before the geared turbine was tried in England. It consisted of—

Two turbo-alternator units, complete with throttle and governor valves.
Four main propelling motors.
Switchboard and control gear, panels, instruments, insulators, &c., and all cables.
Two main motor-driven circulating pumps.
Two main motor-driven air pumps.
Two main motor-driven hotwell pumps.
Four main motor-driven oil pumps.
Two motor-driven oil coolers and tanks.
Motor-driven blowers for ventilating main motors.
Spare parts for the above apparatus.

The total estimated weight of this installation was 700 tons; it actually proved to be 590 tons. The contract price for the above is stated by the General Electric Company to have been $431,000, or, say, £86,500 at par. It is very difficult to compare costs and prices in different countries; but as regards weight, four sets of turbines and gearing with the corresponding auxiliaries would weigh about 470 tons. It was on these broad lines of economy, complication and weight that our article suggested that, for British purposes, electric transmission would prove less suitable than gearing. We may go a little further.

Our electrical contemporary roundly condemned our desire to know more about the performance of these electrically propelled ships, and cites the mass of information published in America. But these data deal almost

entirely with material and not performance in service, which is precisely what is needed. Little or nothing has been published about the Swedish sets, except the comparison between the S. S. *Mjölner* and a sister ship, and we refrained from quoting some of the difficulties which have been experienced in the smaller vessels because they bore in no way on the subject we were discussing. One of the problems in electrical propulsion on any large scale lies in the adequate ventilation of motors and generators. These cannot be ventilated directly with salt-impregnated air, and in the American ships, which require two fans to each motor and generator—some 140 horsepower which does not figure in the above consumptions—special arrangements are necessary for drying the air before it is used. This air supply to the motors might easily be interfered with by damage exterior to the engine-room. Again, the entire control of the whole propelling machinery is centered in the switchboard in the central engine-room, as it must be centered in any electric ship. There is no such concentration of control, no such wholesale reliance on immunity from accident on one item, failure in which means utter breakdown, to be found in any other department. Guns can be fought or the ship can be steered if director tower or conning tower be put out of action. The greatest care has naturally been taken in the design to prevent accident from internal and external causes, but the possibility of all shafts being stopped through a single fault is peculiar to electric transmission, and does not arise in geared installations. The prospect of a bad short circuit in an engine-room is not pleasant to contemplate. The effect of the inevitable condensation of vapor is a further point on which experience requires to be gained. Yet we presume that if we pursue this subject we shall incur the risk of being accused of running down electric propulsion, which is far from our intention. We know that special arrangements were made in the American vessels to guard the cables against this; it is only one of very many problems that require solution before electric transmission can be adopted with the same degree of certainty as the geared turbine is. And the principal one among these for merchant steamers is the question of relative economy and first cost. A secondary one is that of control, and this cannot satisfactorily be accomplished by means other than electrical—for instance, by the use of a clutch—for the reason that the propeller is then not under continuous control as it is in the American ships, or with gearing, even if a satisfactory clutch could be devised.

The extent to which electric propulsion has been adopted, except in the United States Navy, is very limited. In addition to the *New Mexico* and her sister ships *California* and *Tennessee*, the four battleships of the *Maryland* class, of about 33,000 shaft horsepower, the six of the *South Dakota* type, of 60,000 shaft horsepower, and all six of the *Lexington* class of battle-cruiser, of 180,000 shaft horsepower, all with four shafts, are all fitted with this form of transmission, the contractors for the machinery being either the General Electric Company, of Schenectady, or the Westinghouse Company, of Pittsburgh. We cannot altogether acquit our contemporary of denying indisputable facts when it repudiates any suggestion that these installations represent the latest practice in electrical propulsion if only because there is nothing else to compare with them. The United States authorities very naturally compare the performance of their own vessels, and the superior economy of the *New Mexico* over her sister ship *Pennsylvania*, which is fitted with direct connected main turbines and geared cruising turbines for use up to 15 knots, is of the order of 20 per cent for speeds of 19 knots upwards and about 30 per cent at 15 knots. At lower speeds, partly because these geared cruising vessels exhibit very good

economy at very low powers, the fuel saving is very small. Compared with the *Mississippi* fitted with direct drive Curtis turbines and geared cruising turbines, there was a saving in weight of over 200 tons in the *New Mexico*. But these comparisons are with ships which were laid down in 1914-15, and not with all-geared vessels, of which there are no big ship examples in the United States Navy, and which in the British service also show superior economics of 20 to 25 per cent over the last direct driving sets fitted. It is impossible not to admire the work of the General Electric Company in particular, and the ability displayed in solving the many and difficult problems involved in applying efficient variable-speed motors to propeller shafts. A vast amount of care and trouble has been devoted to these designs, and the case for electric propulsion is less likely to be aided by criticism of them by exuberant enthusiasts than by their offering reasonable proof that any other form is likely to meet the drastic requirements of efficiency and reliable operation which are now available in other types of transmission.—"The Engineer," March, 1922.

NODAL ARRANGEMENTS OF GEARED DRIVES.*

By Dr. J. H. Smith.

An inspection of the teeth of gear wheels which have seen service in a faulty arrangement will convince the most optimistic that hammering has been taking place. Pitting is found at the pitch line, rapid wear occurs and grooving of the faces of the teeth appears; flakes and particles of metal are found in the oil drains; the edges of the teeth are sometimes frayed and occasionally the teeth are torn off at their roots and show typical fatigue fractures, thus revealing the presence of fluctuating stresses of range from 30 to 40 tons per square inch; irregularities of pitch sometimes appear. All these items suggest that the teeth have been hammered in a violent and periodic manner. The matter is serious in the case of some double geared ships; there are apparently quite a number running below their economical and designed speed owing either to this trouble or fear of it.

SINGLE AND DOUBLE-REDUCTION GEARS.

Occasionally a double geared ship is produced which appears to give complete satisfaction; occasionally a roarer is turned out which is not satisfactory even at half-speed. The difference in the percentage successes of single and double reduction geared ships appears to be very marked; the single reduction arrangement is considered by many a satisfactory one, but there is a strong feeling amongst engineers that the double reduction arrangement is not. The conditions are so bad in some cases that the performance of double geared ships is below that of ships fitted with reciprocating engines.

Various methods have been suggested for the reduction or elimination of tooth pressure fluctuations and hammering actions, but in all such changes the object in view has evidently been to obtain the effects of more accurate cutting and lower surface pressure. The modern tendency as regards tooth pressure appears to be to return to 500 or 600 pounds per inch width of wheel, a value used by Boulton & Watt for rough cast-iron wheels. It would appear, then, that we are not advancing very rapidly in this branch

* Abstract of paper read at the Spring Meetings of the Sixty-third Session of the Institution of Naval Architects, April 6.

of engineering. As regards the general arrangement of a geared drive, it has been suggested that flywheels and elastic couplings should be added. Elastic shafts of various forms and elastic wheel rims have also been suggested, introduced in some cases with the object of securing a more uniform distribution of load over the width of the wheel, in others with the object of allowing for irregularities in the teeth.

There appear to be only two proposed arrangements in which some slight indication of the introduction of scientific ideas is in evidence. In one of these it is proposed to extend the prime mover shaft beyond its pinion and to add to this extended shaft a flywheel so contrived that the "node" of torsional vibration of the shaft coincides with the pinion; in the other system the main gear wheel is to be placed at the node of torsional vibration of the propeller shaft. We are not informed how these peculiar conditions are to be secured.

INVESTIGATION.

With the object of attempting to obtain a solution of the problem of geared propulsion the writer was invited by Mr. Chas. E. Allan, director of Workman, Clark & Co., Belfast, to carry out an investigation on both experimental and mathematical lines. On taking up this work the writer was brought into touch with an interesting example of a roaring ship. The peculiar behavior of this ship he found could not be explained in any manner by a simple theory, and he was gradually led into a full investigation of the whole question on the broadest possible lines. The main features of the mathematical side of the investigation in their simplest form are recorded in this paper, the results of this investigation leading to what appears to the author to be the only possible solution of the problem.

In order to simplify a proposed elastic shaft geared drive, with the object of securing an arrangement which shall have the least number of possible critical speeds, and therefore the least chance of coincidence of any of these with any impressed periodic actions arising either from the action of unknown internal irregularity or known and unknown external periodic disturbances, a certain amount of ingenuity is required. No common line of procedure can be chosen which will be suitable for the most general arrangement.

When designing a drive we may have to study the peculiarities of the known disturbing elements, and in some cases it may be necessary to split these elements into their component parts. In all such cases the component parts will consist of a fundamental and a series of harmonics having periodicities simple multiples of the periodicities of the main feature of the disturbance. Such considerations will lead to the formation of a family of periodicities. An arrangement of the drive will then have to be made of such a nature that the possible critical speeds are so placed that synchronism does not occur between any one of them and any member of the family of the impressed periodicities. General considerations on a question such as this are, of course, always difficult to deal with, but after carefully considering the possible variations of the numerous elements with which we are concerned in elastic shaft geared drives, the writer is led to the opinion that the required reductions are best secured by:—

(1) Simplifying the system.
(2) Tuning the system.
(3) Making the elasticities and masses of certain shafts dynamically similar.
(4) Adjusting the phases of the disturbances.

(5) Attempting the elimination of high periodicity disturbance by lowering the pitch of the possible free vibrations of the system.
(6) Arranging the periodicities of the disturbance in a definite and simple family.
(7) Reducing the moments of inertia of the wheels of the attendant train to the least possible value.

There may be other items worth considering, but so far the writer has not discovered them.

SIMPLIFICATION OF SYSTEMS.

The difference in behavior of single reduction and double reduction systems would appear to be partly explained as arising from the simple nature of the former. In a single reduction arrangement there are only one-half the number of tooth connections, and hence roughly speaking only one-half the number of possible critical speeds and smaller critical speed ranges than in the corresponding double reduction arrangement. With the present haphazard method of designing, then, the single reduction arrangement has a far better chance of success than the corresponding double reduction type. The suggested use of a flywheel on the propeller shaft is without doubt a mistake; it would increase the degrees of freedom of the system and hence the number of possible critical speeds. Any change in the moments of inertia of the propeller or of the shaft dimensions would not increase the number of possible critical speeds, and is therefore not objectionable. It may be observed also that we must cut out all elastic shafts in the gear box; that is, we must bind the driver and driving wheels mounted on any shaft as rigidly together as possible.

In a tuned system, then, we may say that very great reductions have been made in the possible number of critical speeds. The following table will give some idea of the changes which can be made by tuning alone in a complex system, consisting of n shaft and m toothed connections.

CRITICAL SPEED OF TUNED AND UNTUNED SYSTEM.

Critical conditions.	Number possible in untuned system	Number possible in tuned system
Acceleration of gear wheels	n	1
Free vibrations	n	2
One couple acting on one shaft	$2nm + n$	$4m + 2$
One couple acting on each shaft	$2n^2m + n$	$4nm + 2$

The critical speed ranges are reduced from n to 2 in all cases.

Simplification secured by the adoption of dynamical similarity.—It is often possible to make a group of shafts, together with their attached masses and transmitted torques, dynamically similar, and in such cases the behavior of all the shafts in the group becomes identical and can be treated as one individual shaft. In this way the number of tooth connections are reduced from the number appearing in the group of shafts to the number appearing in one shaft, and hence the critical ranges will be the same for all members of the group. The simplest conditions to be satisfied in such a reduction are first, that the shafts of the group are tuned to the same free periodicity; second, that the masses mounted on the shafts are identical; and third, that the torques transmitted by the shafts are the same. It may here be noted that dynamically similar shafts do not help in simplification unless the system is tuned.

ADJUSTMENT OF PHASE OF DISTURBANCES.

In certain cases the phase of the disturbing periodic couples will be known with a fair degree of accuracy, and hence by an adjustment of various shafts it will be possible to arrange that these couples cancel each other when taken in pairs. In such arrangements the shafts considered will have to be tuned to the same periodicity and assembled in an appropriate manner. As an example of this, we might mention that the simplest nodal arrangement which could be devised, if the elimination of the fluctuation of torque arising from propeller blade disturbance is the important consideration, would consist of twin propeller shafts connected to one gear box and suitably assembled.

Lowering of the pitch of possible free periods.—The author suspects that in many cases angular oscillations have been occurring in turbine rotors and have been so pronounced as to give rise to appreciable fluctuating couples. If such periodic torques are possible it would appear that they can only be high periodicity effects, and can only be avoided by tuning down the possible free vibrations of the system to the lowest pitch.

PERIODICITY OF DISTURBING TORQUES.

In seeking simplification one is naturally tempted to consider not only the method of reducing the number of critical speeds but all other aspects of the question. The periodic disturbing couples, large or small, do not appear to offer anything promising in this direction at first, but a little consideration will show that a simplification is possible here. At the present time there appears to be no attempt made in this direction. How the speed ratios of the various elements of a geared drive are chosen one would have difficulty in finding out. As a matter of fact such a question, in the writer's opinion, is treated in the same way as many others in this branch of engineering. If we consider the simple case of a three-shaft drive, and for simplicity assume that the chief disturbance arises from propeller blade action, it will be seen that the possible periodicities are b N, 2 b N, 3 b N, &c., where b is the number of blades in the propeller and N is the revolutions per minute. These periodicities will be effective, and it is therefore absurd to add further periodicities to this long series. In order, then, to reduce the disturbances to the smallest and simplest possible family, the speeds of rotation of all other elements of the system should be arranged so that they are members of this family and not strangers. That is, the speeds of the primaries and the turbines should belong to this family.

MOMENTS OF INERTIA OF THE WHEELS.

It can be shown that the smaller the moments of inertia of the attendant wheels the more nearly do the various critical speeds merge into one value, and hence the smaller speed range over which these criticals are spread. We may look upon the matter in perhaps a simpler way; for example, in an ordinary three-shaft arrangement the most dangerous elements would appear to be the primaries; when working at or near a critical speed, these wheels are instantaneously free and uncontrolled by elastic connections, and hence may take up independent rebounding hammer motions. If such a thing is possible the only way we can reduce the effect is by reducing the "weight of the hammer," *i. e.*, the moment of inertia of the primary wheel. In the single reduction arrangement the hammer does not exist.

General conclusions on simplification.—We are led, as a result of our investigation, to the conclusion that the tuning or nodalising of any geared drive is the most direct and important dynamical means of attacking the problem when our object is the reduction of the number and magnitude of the possible critical speed ranges.

THREE-SHAFT ARRANGEMENTS.

In order to illustrate the use of the general equations and conclusions deduced, the author deals in a broad way with two simple problems, in which the arrangement of gears and elastic shafts consists of two prime mover shafts geared to one propeller shaft, namely, the common form of arrangement used in ship propulsion. If it was desired to carry our simplification in a haphazard design to the utmost limit, the arrangement would be made to consist of either a single reduction train, or if not possible, then double reduction train with only one prime mover, as in this way the critical speed ranges would be cut down to a minimum. A double reduction arrangement with one prime mover will possess almost all the advantages of a single reduction arrangement with two. In any case we shall reduce our system to a one mass per shaft system, and where double reduction is employed shall employ rigid shaft connections between the primary wheels and pinions, and make the moment of inertia of the primaries of the least possible amount.

CRITICAL SPEEDS.

It is shown that for the simplest possible three-shaft arrangement there are two critical speed ranges, or ten critical speeds when double reduction gearing is employed, and six critical speeds when single reduction gearing is used. The problem then, is reduced to the adjustment of the critical speed ranges, or critical speeds, so that they fall in positions amongst the family of impressed speeds in such a manner that neither coincidence nor approximate coincidence is met with over that part of the speed range which is considered important. This is, in general, a fairly simple thing to do when a design is being prepared *ab initio*, but would appear to require some little patience and ingenuity when one is concerned with the conversion of haphazard arrangements.

It may be observed that in a haphazard arrangement of a three-shaft drive, in which stiff turbine shafts are employed, and hence in which high periodicity criticals may be met, the total possible critical speeds, if it can be maintained that the turbine disturbances are negligible, is 27 in a double reduction and 15 in a single reduction arrangement. Moreover, the critical speed ranges of the tuned system can be determined with a fair degree of accuracy, whereas those of the haphazard system cannot.

Clearly defined critical speeds are apparently seldom met with in haphazard arrangements, and hence the writer is of opinion that there is no clearly defined oscillating system in the majority of cases, but a hopeless jumble of vibrations and chatter to which mathematics cannot be applied. It may be maintained that the high periodicity criticals are outside the working range of the arrangement. The writer cannot see how this can possibly be, as the number of impressed speeds covers such a wide range, and when the harmonics of the propeller's disturbance are considered, such suggestions seem altogether out of the question. The writer has spent a great amount of time in analyzing oscillating systems of the three-shaft kind, and has never yet found one case of haphazard arrangement where all the criticals were not below the speed of revolution of the highest speed turbine shaft.

Critical speeds present no difficulty if they are clearly defined; the sea-going engineer will unconsciously open or close his stop-valve when excessive vibration is met with. If an arrangement of machinery is manufactured which has a multitude of critical speeds within its normal speed range, if a number of these critical speeds occur close together in the working range, adjustment for evasion is impossible.

In a tuned system under certain conditions, generally arranged to correspond with half-speed, clearly defined critical speeds are met with. There is no doubt about the meaning of such to an individual standing on or near the gear case if the periodicity is low, as the phenomena resembles an earthquake and extends over a speed range of from four to six revolutions per minute, but there is no extra noise while working under these conditions. Noise, chatter, hammering and roaring are apparently not to be associated with low periodicity criticals in a system where defined oscillation exists. It appears to the writer that the most dangerous type of critical speed is that of high periodicity. Such criticals are dangerous because we have no clear evidence of their presence except in the rattle and roar, which when once started gradually increases in volume as the speed increases. It was the peculiar behavior of a true roarer which led the writer to suspect that there were other disturbances to be taken into consideration besides those arising from propeller blade action.—"Shipbuilding and Shipping Record," April 13, 1922.

A NEW FORM OF GEAR.

The increasing application of reduction gearing to marine propulsion, particularly during the last decade, has called for machines and processes of greater precision in manufacture. It would appear that the final answer to the question of tooth pressure has not yet been prepared, and while the laboratory determination of what pressure may be economically withstood by the various metals for wheel teeth, the great difficulty of maintaining under marine conditions the contact of wheel and pinion teeth along the whole face must be recognized. If inaccuracies, either in the form of teeth, alignment, or balancing of parts were the sole causes of power transmission troubles, there would be no excuse for their continued existence, but there are and always will be certain varying conditions on land and sea which must be overcome. In a paper recently read before the Institution of Engineers and Shipbuilders in Scotland, Mr. W. Rees Darling described some features of a new autopitch gear which consists essentially of a wheel body with loose pieces of thin plate fitted close together round the rim, forming, as it were, a regular laminated ring in the position usually occupied by teeth, and their corresponding spaces. It is claimed for this gear that a pinion of this construction is able to accommodate itself to any straight tooth in a wheel and will continue to do so until the tooth is entirely worn away. It is therefore independent of contour and thickness of tooth, also the tooth may engage with it at any point of its circumference, and it is thus independent of pitch. There are no tooth clearances, and therefore no backlash. Wear may be taken up by adding new plates, and this pinion may be moved into gear at a speed of 2,000 revolutions per minute without danger or shock. Unfortunately, no data as to efficiency or economy has as yet become available, and the application of the gear to the transmission of high powers such as are met with in marine work is problematical. Nevertheless, the new gear makes such a fundamental departure from existing theory and practice as to be worthy of some attention.—"Shipbuilding and Shipping Record," March 30, 1922.

THE PROTECTION OF CONDENSER TUBES FROM CORROSION AND SEASON CRACKING.

Causes of Corrosion and Methods of Prevention—Composition of Tubes—Tube Treatment.

Few subjects have more consistently engaged the attention of operating engineers ashore and afloat or have figured more prominently in the papers and discussions of technical institutions and research committees than the effective protection of condenser tubes against corrosion and the several other factors which contribute towards failure. At the annual general meeting of the Institute of Metals, held on March 8, two further papers on this somewhat complex and many-sided subject were read, as follows:—"Notes on the Corrosion and Protection of Condenser Tubes," by C. D. Bengough, M. A., D. Sc., and "Further Studies in Season-Cracking and its Prevention—Condenser Tubes," by H. Moore, O. B. E., Ph. D., F. I. C., and S. Beckinsale, B. Sc.

As investigator to the Corrosion Research Committee, Dr. Bengough has carried out scientific investigations covering every aspect of the problem during a period of approximately 10 years, and definite conclusions have been put forward based on an extended experience of the practical working of condensers. As a strictly practical paper, taking the fullest account of the respective viewpoints of the manufacturer of the tubes, the maker of the condenser, and the operating engineer, this may rightly be said to be the most authoritative expression of opinion on the methods of protecting condenser tubes from corrosion that it is possible to prepare at the present time. While appreciating that the most fertile cause of failure of condenser tubes in service is corrosion, the authors of the latter paper point out that occasional failures occur, both in store and on service, by the development of cracks arising from internal stress, and this paper is a record of the work done in consequence of a suggestion made by the President, Engineer Vice-Admiral Sir George Goodwin, in the discussion on a paper presented to the Institute two years ago by the same authors. At that time the President suggested the application of low-temperature annealing to Admiralty condenser tubes as a possible means of reducing or eliminating the internal stress in brass.

LOW-TEMPERATURE ANNEALING.

Previous investigation has shown that the origin of internal stress in drawn brass tubes is attributable to drawing through a die without internal support, and tubes in which the internal diameter has been reduced in this way without any appreciable reduction of wall thickness are described as "hollow-sunk." Primarily the experimental work under this heading was directed to the subject of the reduction of internal stress by low-temperature annealing and the effect of such treatments on the strength and range of elasticity of condenser tubes. The ground covered by this work comprised the heat treatment of spun caps of highly stressed condenser tubes; of strips of condenser-tube brass maintained in a form causing a known initial stress; and of tensile test pieces of the same material.

The experiments on the low-temperature annealing of internally stressed condenser tubes and spun caps agreed with the quantitative results obtained by the work on elastically bent strips of Admiralty condenser-tube brass, in which strips cold-rolled to 120 and 160 Brinell hardness, initially stressed

to 10, 15 and 20 tons per square inch maximum tension at the surface, were annealed at temperatures in the range 200 degrees to 325 degrees C. The results obtained may be summarized as follows:—

1. At the lower temperatures the rate of reduction of stress is fairly rapid at first, but becomes slow when the stress has been substantially reduced. Important stresses remain after six hours at 200 degrees or 225 degrees C.

2. As the temperature is raised the rate of reduction of stress increases, but shows the same characteristic of slowing down as the stress falls during the treatment.

3. The higher the initial stress the higher is the remaining stress after a given treatment in condenser-tube brass of the same hardness, although the amount of stress removed is greater the higher the initial stress.

4. The higher the hardness of the brass, the lower is the remaining stress after a given treatment and for a given initial stress.

5. A large reduction in the amount of stress is brought about by annealing conditions (temperature and time) which raise slightly the hardness of the cold-worked material, and the stress can be reduced to a low value at which it appears to be innocuous, by treatments which do not appreciably lower the hardness and strength.

Although it is generally believed that season-cracking may be brought about in brass solely by the action of internal stress this view is contested by the authors, who claim that chemical or other action is contributory, and they hope to establish this claim as the result of further experiments which are at present in progress.

CORROSION—ITS CAUSES.

The factors contributing to the failure of a condenser tube are so many and varied that it is rarely possible to point to any single factor as being the sole cause of actual perforation, and when such perforation has been observed it will more generally be found to have been the cumulative result of several different factors. Moreover, it is doubtful if similar conditions ever obtain in any two condensers, and while experiments carried out in any individual plant or ship may have been attended with marked success, such experiments cannot be applied in another case with any degree of hopefulness unless all the conditions are known.

The various causes of tube failure may conveniently be classified under two heads, viz., (1) Causes due to the tubes, (2) Causes due to external conditions.

(1) *Causes due to the Tubes.*

(*a*) Laminations, spills, &c.
(*b*) Faulty composition.
(*c*) Unsuitable crystalline structure.
(*d*) Mechanical failure, *e.g.*, collapse under packing, splitting, season-cracking, &c.

(2) *Causes due to external conditions.*

(*e*) Local accumulation of corrosion products.
(*f*) Settling of foreign substances on the tubes, sometimes giving rise to (*e*).
(*g*) Subsidiary supplies of steam entering condenser apart from main, giving high local temperature, and, sometimes, external erosion.
(*h*) Inlet deterioration on upper internal surface of tube.
(*j*) Electrical leaks and thermo-electric action.

(*k*) Water speed, especially with dirty inlet water, as in certain harbors and docks.

(*l*) Entrance into certain waterways, especially when tubes are new, *e. g.*, ship canal and certain tropical rivers; certain docks; the Tyne at certain periods of the year; the upper reaches of the Clyde.

(*m*) Air-sucking due to rolling in bad weather, air leaks in pump line, &c.

(*n*) Unsuitable design of water-end.

(*o*) Ferrule troubles.

(*p*) Too high a temperature.

(*q*) The presence on new tubes of certain iron compounds.

(*r*) "Short circuiting" of contaminated cooling water.

(*s*) Seasonal or other variations in water composition.

Corrosion has been more generally found to take place on the inner or water side of the tube and may take one or two forms to which the following names have been applied:—

(1) "Complete corrosion," *i. e.*, both copper and zinc are attacked, and either removed in solution or converted into basic salts.

(2) "Selective action" or "dezincification;" apparently only the zinc is attacked and the copper left *in situ*.

The former is usually characterized by the presence of a considerable amount of blue-green basic salt on the inner tube surface. This salt colors the white salt of zinc, and of red cuprous oxide; a proportion of the zinc passes into solution. Dezincification is generally characterized by a white salt of zinc overlying considerable deposits of metallic copper.

COMPOSITION OF TUBES.

Laminations and spills are usually formed during the drawing process, and while these may not be in themselves the initial causes of local action, their presence, when corrosion has commenced from other causes, is likely to accelerate disintegration. Specifications covering the composition of condenser tubes are many and varied. Generally it may be said that for an ordinary 70:30 brass tube the copper element should fall between the limits 69.5 and 71 per cent; zinc should not be more than 30 per cent; and, unless added for a specific purpose, other metals should not exceed 0.4 per cent. The presence of iron in excess of 0.1 per cent is undesirable and will usually be found to accelerate the white salt attack in sea and estuarine waters. Nickel in limited quantity has given beneficial results, but on no account should aluminium be present. Tin in proportions up to 1.0 per cent does not influence the behavior of the tube as regards corrosion, but the presence of lead will be found to increase the tendency to dezincify. Admiralty tubes contain 70 per cent copper, 29 per cent zinc, and 1 per cent tin, and this composition should generally be adopted in preference to the plain 70:30 brass unless with waters which are known to be only very slightly corrosive. Tubes which have received a final annealing treatment in the temperature range 350 degrees C. ± 25 degrees C. in an oxidizing atmosphere have a fine crystalline structure and this treatment has no appreciable softening effect on the tube, but will be found to retard corrosion to some extent. Mechanical failures are the result of the tube being either too soft or too hard, and these failures may be avoided by careful treatment in the drawing shop. A soft tube will bulge under ferrule pressure and a hard tube may fail by splitting. Season cracking is a common form of mechanical failure and has already been dealt with.

The chemical changes involved by the local accumulation of corrosion products are somewhat complex and an attack due to salt accumulations is entirely a matter of chance and is obviously beyond the control either of the tube manufacturer or the operating engineer. Similarly the presence of foreign substances such as weed, clinker, wood and sand is equally outwith the control of both parties. These substances, while chemically harmless in themselves, are nevertheless instrumental in collecting active factors of corrosion. With new tubes which have assumed no protective scale the danger from this source is greatest. When auxiliary steam is led into a condenser at such a point that it impinges on the lower cold tubes external erosion may be set up and internal corrosion has also been observed due to the high temperature. Such auxiliary steam should be introduced at the top of the condenser and as far removed from the tubes as the design may permit. Electrical leaks are rarely productive of corrosion, but when such action does take place it will be found to develop at or near the tube ends. Water speed presents one of the most complex phenomena in the subject of corrosion. A local increase in water speed, particularly with sea-water, is productive of the condition known as "foaming," and this condition may be aggravated by the presence of air. The speed near the tube walls is of great importance and generally it may be said that increased speed means increased corrosion. Foaming has been observed with sea-water at speeds of 5 to 6 feet per second and has been eliminated by reducing the speed to 2 to 3 feet per second. Restricted water-ends have been found to set up undue agitation in the water with consequent foaming and corrosion. Ferrules of a composition 60:40 brass are more sensitive to corrosion than 70:30 brass, and it is frequently desirable to use an alloy of 70:29:1. Leaky packing and the presence of clinker, wood, &c., are also fertile sources of ferrule corrosion. Excessive temperature of the cooling water, especially in the tropics, is a source of trouble and generally it may be said that within limits corrosion increases directly as the temperature. Short circuiting is of frequent occurrence in the case of works situated on river sides. The water in the outtake jack-well may become impregnated with factory products and find its way to the intake jack-well at certain states of the tide.

CORROSION—METHODS OF PREVENTION.

The various methods recommended for the lessening of corrosion may conveniently be resolved under two heads, viz.:—(1) Water treatment; (2) Tube treatment. Physical treatment of water, *e.g.*, by filtration, is not practicable on a large scale, unless in the case of make-up water for cooling ponds, and need not be here discussed. Several methods of chemical treatment, such as softening, neutralization, &c., have occasionally been adopted, but as this is distinctly outside the province of the operating engineer it is of little practical interest. Of the several methods of water treatment the electrolytic method has been most generally used and has given the most practically successful results. In its simplest form it consists of zinc or steel slabs placed in the water-ends, connected electrically to the tube plate or condenser body forming an electric couple with the tube to be protected. Although higher than steel in the electrochemical series zinc is much inferior for purposes of protection. Primarily it is more expensive and is liable to reversal of polarity; furthermore it gives rise to the formation of corrosive products which may reach the tubes, and it is also subject to exfoliation loss. Unless care and judgment are exercised in the use of steel blocks little or no protective value will result. As many blocks should

be used in the water-ends as is possible without interfering with the circulation and these should be disposed symmetrically opposite and parallel to the tube plate. If short circuiting is to be avoided the faces of the blocks should be nearer to the tube plate than to any part of the water-end. Perfect metallic contact between the blocks and the tube-plate and between the tubes and the tube-plate should be ensured and in the case of the tubes a soft metal ring should be inserted when packing. Protection afforded by this method has been found to extend not more than 18 inches from the end of the tube. By the application of heavier currents through the medium of a dynamo or secondary battery kept at a constant value of about 3½ to 5 ampères per 1,000 square feet of protected surface good results have been obtained, but this method is not only much more expensive, but also calls for very specialized attention to details of installation and upkeep. The positive pole of the dynamo or battery is connected to the steel block or anode, and the negative pole to the tube plate. Contrary to the general belief electrolytic protection does not eliminate corrosion, but merely at best retards it.

TUBE TREATMENT.

Tube-cleaning by means of "rodding out" with a view to removing such foreign deposits as clinker, shells, wood, &c., has been attended by good results. Limited space frequently renders the "rodding out" process impracticable and an alternative method of cleaning is by the use of acids. An objection to the use of acids is that while it may remove the harmful deposits which set up local action, it also has been found to disintegrate the protective scale. A new tube surface is thus exposed which may increase the danger of local action beginning. Objection has usually been made that the direct attack of the acid on the metal of the tube would be serious, but extended experiments have shown that, provided cold acid is used and of a concentration just sufficient to attack ordinary scales this factor is not serious, inasmuch as the acid treatment is not prolonged.

New tubes have been found to be particularly susceptible to corrosive action and the preparation of artificial scale prior to tubes being used has been attempted but with only partial success. The principal difficulty has been found to be the time factor, as experience shows that to obtain a really resistant scale its deposition must be effected slowly. As a period of several months has been suggested, this method has certain commercial disadvantages, but the results obtained are such as to encourage the further study of this phase of the subject.

There are those who today advocate the lining of condenser tubes with metallic layers, and it is known that during the recent war tubes lined with lead were used by the German fleet, with, it is understood, quite satisfactory results. For adequate protection the layer of lead should have a certain minimum thickness, and here again a secondary difficulty arises, since very thin layers which appear continuous are liable to develop porous conditions. Such conditions permit the access of water between the layers and chemical action results. Lead lining should only be used where low water speeds are possible and when accurate manufacturing conditions are available.

In the choice of a suitable condenser tube the nature of the water supply should, wherever possible, be definitely ascertained and fully taken into account. To classify the several industrial waters is outwith the scope of this article. Within the limits allowed by first cost, the use of nickel has given good results where chloride waters have been encountered. Monel metal is believed to contain the highest proportion of nickel found in any condenser tube alloy, and as a resistant to the corrosive action of acid

waters Monel metal has given satisfactory results. Aluminium-copper tubes containing as much as 8 per cent aluminium have frequently been tried. Generally, in practice, these tubes are unreliable and should be avoided. For ordinary conditions met with in everyday practice the use of Admiralty alloy 70:29:1 is to be recommended for condenser tubes and ferrules, as this alloy has been found to give the most consistently reliable results.—"Shipbuilding and Shipping Record," March, 30, 1922.

ELECTRO-DEPOSITION OF IRON.

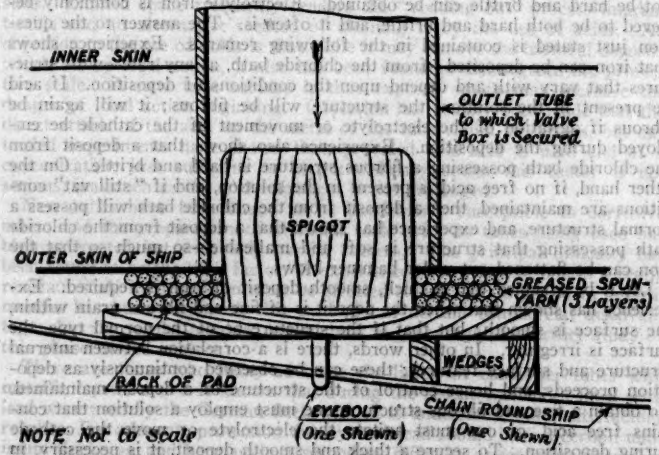
A bulletin just issued by the Department of Scientific and Industrial Research gives an account of an extensive research carried out by Mr. W. E. Hughes, late chief research chemist, Electro-Metallurgical Committee, Ministry of Munitions, into the electro-deposition of iron. Mr. Hughes comments on the similarity between the processes of crystallization and deposition and gives the following practical hints.

He has, he says, been asked several times how a deposit of iron that will not be hard and brittle can be obtained. Electrolytic iron is commonly believed to be both hard and brittle, and it often is. The answer to the question just stated is contained in the following remarks. Experience shows that iron can be deposited—from the chloride bath, at any rate—with structures that vary with and depend upon the conditions of deposition. If acid be present in the solution, the structure will be fibrous; it will again be fibrous if agitation of the electrolyte or movement of the cathode be employed during the deposition. Experience also shows that a deposit from the chloride bath possessing a fibrous structure is hard and brittle. On the other hand, if no free acid is present in the solution, and if "still vat" conditions are maintained, then a deposit from the chloride bath will possess a normal structure, and experience has shown that a deposit from the chloride bath possessing that structure is soft and malleable—so much so that the iron can be flattened out under hammer blows.

Let it be assumed that a thick, smooth deposit of iron is required. Experience has shown that when the deposit is of fine and fibrous grain within, the surface is smooth; but that if the structure be of the normal type, the surface is irregular. In other words, there is a correlation between internal structure and surface features; these can be observed continuously as deposition proceeds, and hence control of the structure of a deposit maintained. To obtain a fine and fibrous structure, one must employ a solution that contains free acid, or one must agitate the electrolyte or move the cathode during deposition. To secure a thick and smooth deposit, it is necessary, in addition, to maintain approximately constant the concentration of metal in the solution; for concentration is always an important, Mr. Hughes believes it to be the dominant, factor determining the structure of a deposit, the other factors affecting the structure, if not perhaps wholly, then, at any rate, especially, in consequence of their effect upon the concentration. To obtain a thick, smooth deposit of iron, one must therefore use a bath containing free acid or must agitate the electrolyte or move the cathode during the deposition, and one must, in addition, see to it that the metal concentration of the bath does not become appreciably diminished.

EXAMINATION OF UNDERWATER FITTINGS WITHOUT DRY-DOCKING.

Before a certain steamer was undocked the underwater fittings were thoroughly examined, overhauled and refitted; and all underwater valves were closed. After the vessel was floated instructions were given for the main inlet and discharge valves to be opened in order to flood the condenser. The discharge valve jammed half open. Divers were sent down but could discover nothing wrong. It had been considered that if the main circulator were started up, the flow of water might dislodge the obstruction. As no steam was available the steam valve chest cover was removed from the circulating engine, and air supply from the shipyard was introduced by means of an adaptor. In this way an air pressure of 80 pounds per square inch was obtained at the engine, and this pressure was found sufficient to run the circulator. This method, however, proved unsucessful, and the discharge valve remained jammed half open. As the cost of redocking the ship was likely to prove a serious item, it was decided to make a plug pad to cover the outlet tube, as shown in the diagram. It was considered that the pressure of the water outside would keep the plug in place during the removal and examination of valve and box.



Method of Plugging Discharge Valve Orifice.

The discharge valve was of the ordinary flat shutter type, and the outlet orifice of tube connecting the inner and outer skins of the ship was 25 inches diameter. Unfortunately, the bilge keel was situated near the discharge orifice and it was not possible to pull the plug up hard to its seat. The plug was constructed as follows:—Two-inch planking was used for the outer cap, and to this a built-up wooden spigot was secured, the spigot being a very slack fit in the discharge tube. The length of spigot was 15 inches. Well-greased spun yarn was wound round the spigot at the base making three layers at the outer flange, as shown in sketch. Divers were then sent

down to place the pad in position. The pad was provided with two eye-bolts, through which chains were passed and these chains were secured to the upper deck on either side, passing entirely round the ship's bottom. As the bilge-keel prevented the chains keeping an even pressure on the pad, wooden wedges were driven between the chains and the pad on the side on which the bilge-keel projected. As no steam was available to work the fire and bilge pump, a connection was made and compressed air was supplied to the steam valve chest of this engine. It was necessary to use this pump in order to keep back any leak into the bilges, and also to keep the discharge tube clear of water when valve had been removed. In addition, a salvage steamer stood by the ship in case of emergency.

When all was ready the valve box cover was removed, and the valve and box examined. It was found that a wooden wedge had worked right into the valve box, turned end on, and so prevented the valve closing. It is considered that one of the dock shore wedges had become held up in the tube, and when the valve was partly opened, had worked its way in as described. The wedge was removed and valve box closed up, the whole operation after fitting the pad having taken less than 10 hours. By this method of repair re-docking was avoided and a saving of several hundred pounds was effected.—"The Marine Engineer and Naval Architect," June, 1922.

THE LATEST SPERRY GYRO-STABILIZER INSTALLATION.

NEW 120-TON STABILIZER TO BE INSTALLED ON AMERICAN PASSENGER LINER "HAWKEYE STATE"—GENERAL PRINCIPLES—SOME PARTICULARS OF ROLLING EXPERIMENTS RECENTLY CARRIED OUT ON AN ITALIAN DESTROYER.

From time to time a great variety of devices has been introduced for the prevention of rolling of ships at sea. Remarkable advances have been made in recent years in the matter of luxurious accommodation on board the largest liners, but it is significant that corresponding progress has not been made in eliminating the very obvious discomforts associated with the heavy rolling experienced in rough weather. That this is so, and that even the largest liners are not proof against the forces of the elements, was demonstrated some months ago when the *Olympic*, on a voyage from New York to Southampton, developed an alarming roll and had several ports stove in during a gale of unusual severity. These anti-rolling devices have assumed various forms. An arrangement of pendulum weights has been tried on a small scale, but owing to the space required and the difficulty of adapting this arrangement to vessels of large size, it has not found extended application. Several well-known liners have been fitted with a system of anti-rolling tanks, and while satisfactory results have in some cases been recorded, the adoption of this system has not been by any means general. The application of the principle of the gyroscope to the stabilizing of ships bids fair to ensure a greater degree of comfort to passengers in all weathers than any device hitherto evolved. The gyroscope as a ship stabilizer has been perfected by Dr. Elmer A. Sperry, and several vessels have already been equipped with the Sperry Gyro-Stabilizer and have given very satisfactory performances.

Among the vessels so equipped may be mentioned the yachts *Naomi*, *Guinivere* and *Lyndonia*, the *Guglielmo Pepe*, an Italian destroyer of 1,500 tons, and the American freighter *Huron*. In addition, there are under construction equipments for two train ferry-steamers for the Japanese Government, a 10,000-ton cruiser for the Japanese Imperial Navy, and a set for

the 20,000-ton United States Shipping Board passenger liner *Hawkeye State*. This represents the largest gyroscope so far constructed in the world, and it is also the first gyroscope to be installed on a passenger liner. A series of tests was recently carried out on this unit at the Philadelphia works of the Westinghouse Electric Manufacturing Company before a representative body of naval and merchant-service engineers. These tests are reported to have been of a highly satisfactory nature, and it is understood that this gyroscope will be installed on the vessel as soon as service conditions permit —probably during this month. The stabilizer weighs no less than 120 tons and is designed for a speed of 800 revolutions per minute. During the test, an overspeed of 10 per cent was reached, *i.e.*, 880 revolutions per minute, and it is stated that such is the energy stored in the flywheel that it will revolve for a period of about 18 hours after the power is shut off.

We produce a diagram illustrating the principle on which the main gyroscope functions. The action is controlled by a small gyroscope situate at a convenient point in the vessel. This control gyroscope registers the direction of the roll and automatically sets in motion the main gyroscope in the direction necessary to counteract the motion of the vessel. The stabilizer consists of a heavy wheel electrically driven and mounted in a cast-steel case which is provided with bearings and is fitted in the ship with the axis of these bearings in an athwartship plane. The spinning axis of the gyro-wheel is at right angles to these bearings. Should the case be moved in its bearings a powerful gyroscopic reaction is produced in the 'thwartship direction. The control gyro, on account of its great sensitiveness, responds to the slightest acceleration of the vessel in the 'thwartship direction and immediately an impulse is received from a wave, the control-gyro energizes an electric motor, known as the precession motor. This motor turns, or precesses, the main gyro-case by means of the toothed quadrant shown in the left of the photograph. Precession of the main gyro-case produces a powerful gyroscopic reaction in the 'thwartship plane, which tends to neutralize the rolling impulse caused by the wave. When this impulse has been destroyed, the control-gyro stops the precession motor and the apparatus is then in a position to deal with the next wave impulse.

A series of rolling curves were recorded from actual tests carried out on the Italian Navy Scout *Guglielmo Pepe* in the Gulf of Spezia from March 3 to 11 of this year. These curves are of considerable interest, and, with the amplitude of roll plotted against a time base the results are readily visible. In the top set of curves the vessel was stopped and was rolling under the influence of a light swell on the quarter. The remarkable roll-quenching effect when the Sperry gyro-stabilizer is brought into action is clearly shown. The two lower curves show a comparison between natural and forced dying-down rolling. In this case the vessel was in still water and rolling was developed by men, the mean period of roll being 9.6 seconds and the maximum amplitude 15 degrees. Under natural conditions dying-down was not complete until 31 oscillations had taken place subsequent to the cessation of the rolling effort. With the gyro in action, quenching was complete within the period occupied by just over one oscillation.

Having regard to the enormous energy generated in a flywheel of this magnitude and revolving at such high speed, considerable discussion has naturally been raised as to whether the gyro-stabilizer does not impose excessive stresses on the structure of the ship. A sea wave approximates in form the trochoidal wave, with a height varying from 1/16 to 1/25 of its length. The maximum inclination of such waves is about 12 degrees but, as this inclination is limited in extent, the effective inclination—inasmuch as this more nearly determines the motion of the ship—should be considered,

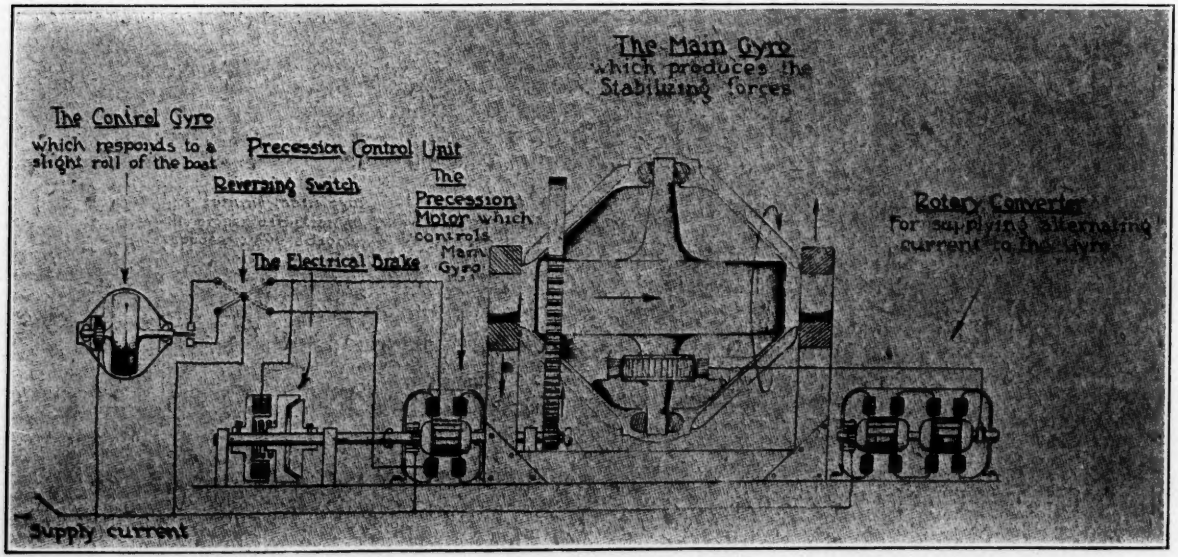


DIAGRAM ILLUSTRATING THE PRINCIPLE ON WHICH THE GYRO-STABILIZER FUNCTIONS.

and this may reasonably be taken as about 3 degrees. The rolling of a vessel is not the direct result of any individual wave, but is the cumulative result of a large number of relatively small impulses. Should the vessel be free to respond to these impulses a heavy roll will inevitably develop, but if each individual impulse is neutralized or quenched as it occurs, the configuration of the vessel will be practically unchanged. The forces brought to bear on a ship by the stabilizer are therefore only those sufficient to quench the roll increment for which the stabilizer may be designed.

Owing to the localized weight and the anti-rolling forces developed, specially reinforced foundations and heavier scantlings in the vicinity are naturally called for, but inasmuch as similarly-strengthened seatings are equally necessary for engines and reduction-gear casings, no undue difficulties should arise in this respect. The loss in deadweight-carrying capacity which might be argued as an objection to the fitting of a gyro-stabilizer, does not materially arise in a passenger ship, and will certainly be completely outweighed by the additional comfort afforded the passengers.—"The Marine Engineer and Naval Architect," June, 1922.

NOTICE.

A NATIONAL ENGINEERING MUSEUM.

After the DeLamater-Ericsson Commemoration on March 9th, the 60th Anniversary of the Battle between the *Monitor* and *Merrimac*, the American Society of Mechanical Engineers appointed a Committee to co-operate with the Committee in charge in making a collection of historical material connected with Mr. Cornelius H. DeLamater and Capt. John Ericsson during their 50 years' association (1840-1890) as the leading factors in the DeLamater Iron Works, at the foot of West 13th St., New York City. These Works were the largest institution of their kind in their day and there during the period mentioned developments took place in Naval-Merchant-Marine, Ordnance and Industrial Engineering which helped materially to raise this nation from comparative unimportance to its recognition as the leading nation of the world, and yet of the details of this accomplishment there is no record. In fact there is nowhere a tangible record of any of the steps whereby this country has attained its present commanding position in the world.

In seeking for a permanent depository for the DeLamater-Ericsson Historical Collection the Committee conferred with the Smithsonian Institution at Washington and its Secretary sent with his reply his latest report on the National Museum which is under his custody. The concluding paragraph of this report reads as follows:

"Museums devoted to history, art and the natural sciences are established in all of the larger nations of the world and many of them have established in addition, educational museums of engineering and industries. Thus England has her

South Kensington Museum: France her Conservatoire des Arts et Matiors (at Paris) and Germany her Deutsches Museum (at Munich) but nowhere in this Nation of ours, the most advanced in the application of the engineering and mechanical arts, is there a similar institution. The commanding place in the world which the United States has reached in the short space of seventy-five years is due largely to the full development and utilization of mechanical power in the exploitation of her natural resources. It is this that has made it possible for the people of the United States to enjoy a standard of living far and above that under which the peoples of the rest of the world exist and still no public sign of appreciation either national or otherwise is to be found anywhere. What more suitable monument could there be therefore than a Museum of Engineering and where could there be found a more logical place for it than as a part of the great National Museum?"

The Committee has responded that it realizes the responsibilities to be incurred in assuming the performance of such a task as has been proposed, but feeling that now is the time to do it, for every day passing is one during which the records needed are being lost and men who were leaders in events are dying off, and feeling that with the backing of the American Society of Mechanical Engineers, the largest Engineering Society in the World, it is strong enough to do it, it has started the movement.

It is desired to make this a national movement in which every engineer in the country will participate. Engineering societies and Alumni Associations of Engineering Schools are requested to communicate for further information with the

JOINT COMMITTEE ON THE
NATIONAL ENGINEERING MUSEUM,
Engineering Societies Building,
29 W. 39th St., New York City.

ASSOCIATION NOTES.

REVISION OF BY-LAWS.

On July 8, 1922, ballots cast for and against a revision of the By-Laws of the Society submitted to the Membership on April 8, 1922, were counted. The vote was as follows:

Total number of votes cast, 233—

For Revision	221
Against Revision	4
No Preference	8

MEMBERSHIP.

The following members and Associates have joined the Society since the last report:

MEMBERS.

Anderson, William B., Lieutenant, U. S. Navy.

Ault, William B., Ensign, U. S. Navy.

Barrett, John B., Lieutenant, U. S. Navy.

Baylis, John S., Lieutenant, U. S. C. G.

Beard, William K., 824 Park Avenue, Richmond, Va., Ex-Lieutenant, U. S. Navy.

Beckley, Chester A., Ex-Coast Guard Lieutenant, 313 Pequot Avenue, New London, Conn.

Benoist, Wilson A., Ensign, U. S. Navy.

Buchanan, Walter B., Lieutenant, U. S. Navy.

Camden, Bernard H., Lieutenant Commander, U. S. Navy.

Christensen, Bernhard, Ensign, U. S. Navy.

Clarke, Reuben R., Lieutenant (CC), U. S. Navy.

Coleman, Beverly M., Ensign, U. S. Navy.

Cotter, James J., Lieutenant, U. S. Navy.

Davis, Edwin W., Lieutenant, U. S. Navy.

Dorry, John E., Lieutenant Commander, U. S. C. G.
Eggleston, Thomas J., Ensign, U. S. Navy.
Ellerton, George C., Lieutenant, U. S. Navy.
Farwell, Lorenzo C., Lieutenant (E), U. S. C. G.
Ferris, Floyd, Ensign, U. S. Navy.
Flanagan, C., Lieutenant, U. S. Navy.
Gibson, Francis S., Lieutenant, U. S. Navy.
Green, Nathan, Ensign, U. S. Navy.
Greenslade, John W., Captain, U. S. Navy.
Grubham, Myron T., Ensign, U. S. Navy.
Hall, Carroll M., Lieutenant, U. S. Navy.
Halverson, Chris., Ensign, U. S. Navy.
Hammes, Roman B., Lieutenant Commander, U. S. Navy.
Hampson, Edward W., Ensign, U. S. Navy.
Hancock, Lewis, Jr., Lieutenant Commander, U. S. Navy.
Hand, Earl V., Lieutenant, U. S. Navy.
Hannah, Tony L., Lieutenant (CC), U. S. Navy.
Hardie, William C., Lieutenant (CC), U. S. Navy.
Harrington, Archibald C., Ensign, U. S. Navy.
Haven, Hugh E., Ensign, U. S. Navy.
Hibbard, Charles, Lieutenant (CC), U. S. Navy.
Hunt, Walter M., Captain, U. S. Navy.
Hutchinson, Myron W., Jr., Lieutenant, U. S. Navy.
Jacobsen, Jenis V., Lieutenant, U. S. Navy.
James, William, Lieutenant, U. S. Navy.
Johnson, Charles M., Lieutenant, U. S. Navy.
Kerr, Raymond E., Lieutenant, U. S. Navy.
Keyes, Frederick C., Lieutenant, U. S. Navy.
King, Frank Robert, Lieutenant, U. S. Navy, Ret.
Kitchin, Howard W., Lieutenant, U. S. Navy.
Lee, Samuel E., Ensign, U. S. Navy.
Lewis, Clyde L., Lieutenant, U. S. Navy.
Lonnquest, Theodore C., Lieutenant, U. S. Navy.
Lutken, Harold I., Lieutenant, U. S. Navy.
McCord, Charles G., Lieutenant Commander, U. S. Navy.

McNair, Frederick V., Commander, U. S. Navy.
Mack, Brice H., Lieutenant, U. S. Navy.
Mead, Stuart P., Lieutenant (CC), U. S. Navy.
Molloy, Thomas M., Lieutenant Commander, U. S. C. G.
Murphy, John J., Lieutenant, U. S. Navy.
Murray, James D., Lieutenant, U. S. Navy.
Nelson, Frank E., Lieutenant, U. S. Navy.
O'Brien, Thomas F., Lieutenant (CC), U. S. Navy.
Odlin, Lawrence A., Lieutenant Commander (SC), U. S. Navy.
Ouellet, Joseph A., Ensign, U. S. Navy.
Pendleton, Perley E., Ensign, U. S. Navy.
Perry, Emil B., Lieutenant, U. S. Navy.
Prosser, Albert L., Ensign, U. S. Navy.
Pursell, Ion, Lieutenant, U. S. Navy.
Quynn, Russell H., Ensign, U. S. Navy.
Ransehousen, Rogers S., Ensign, U. S. Navy.
Risser, Frank, Lieutenant, U. S. Navy.
Roach, Philip F., Lieutenant, U. S. C. G.
Robinson, Elbert B., Lieutenant, U. S. Navy.
Robnett, John D., Commander (SC), U. S. Navy.
Royal, Forrest B., Lieutenant, U. S. Navy.
Ryan, James B., Lieutenant, U. S. Navy.
Schwab, Frederick, Lieutenant (SC), U. S. Navy.
Scott, Alfred G., Ensign, U. S. Navy.
Schackford, Chauncey, Captain, U. S. Navy.
Shannon, James, Lieutenant, U. S. Navy.
Sheard, Walter G., Captain, U. S. M. C.
Sheehan, John M., Lieutenant, U. S. Navy.
Shugg, Carleton, Ensign, U. S. Navy.
Sinclair, Hugh, Lieutenant, U. S. Navy.
Smith, H. Donald, Ensign, U. S. Navy.
Smith, William E., Ensign, U. S. Navy.
Staton, Adolph, Commander, U. S. Navy.
Sterling, Morton B., Ensign, U. S. Navy.

Stuart, Alexander, Lieutenant, U. S. Navy.
Styer, Charles W., Lieutenant, U. S. Navy.
Sullivan, William D., Lieutenant, U. S. Navy.
Sweeney, Ralph D. F., Ensign, U. S. Navy.
Thompson, Robert R., Lieutenant, U. S. Navy.
Thomson, Gerald W., Lieutenant (CC), U. S. Navy.
Wade, William C., Lieutenant, U. S. Navy.
Wainwright, Dallas B., Lieutenant Commander (SC), U. S. Navy.
Walker, Robert J., Lieutenant, U. S. Navy.
Wartman, Samuel L., Lieutenant, U. S. Navy.
Weed, Walter H., Jr., Ensign, U. S. Navy.
Whitman, Ralph, Commander (CEC), U. S. Navy.
Williamson, Tom H., Ensign, U. S. Navy.
Wilson, Eugene E., Lieutenant Commander, U. S. Navy.
Woodson, Eugene M., Lieutenant Commander, U. S. Navy.
Wright, Henry, Captain (CC), U. S. Navy.
Zinn, Ralph T., Ensign, U. S. Navy.

ASSOCIATES.

Howarth, H. A. Stevens, General Manager and Chief Engineer, Kingsbury Machine Works, 4836 Roosevelt Boulevard, Philadelphia, Pa.

Nibbs, E., Electric Boat Co., Groton, Conn.

Miller, Walter M., Mfg. Mgr., Marland Refining Co., Ponca City, Okla.

Parks, Harry S., Vice-President, Pneumercator Co., Inc., 4410 Osage Avenue, Philadelphia, Pa.

Wellman, Jarvis B., 788 W. 48th Street, Norfolk, Va.

ADDRESSES.

To insure prompt delivery of your JOURNAL, please do not neglect to keep the Secretary-Treasurer informed of your correct address.

ERRATA.

Dr. Edgar Buckingham and Mr. C. F. Bailey have been appointed Civil Members of the Council, and Commander Q. B. Newman, U. S. C. G., has been appointed as Naval Member of the Council, vice Lieutenant M. R. Daniels, U. S. C. G., resigned.